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THE UNIVERSITY OF ALBERTA

HEAT EXCHANGERS IN ANIMAL CONFINEMENT HOUSING

by

© DONALD A. KENNEDY

A THESIS
SUBMITTED TO THE FACULTY OF GRADUATE STUDIES AND RESEARCH
IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE
OF MASTER OF SCIENCE

DEPARTMENT OF AGRICULTURAL ENGINEERING

EDMONTON, ALBERTA
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Date. July 12, 1988

ABSTRACT

Heat exchangers can be used in animal confinement housing to recover energy lost through ventilation. Previous work on this topic was studied with a focus on problems encountered with past installations. Optimum heat exchanger performance properties were investigated and a parallel-flow prototype was designed and fabricated using the results obtained. Systems were developed for measuring and describing heat exchanger performance. Once fabricated, the prototype heat exchanger was installed in a 7000 bird broiler barn and tested over two growth cycles. The results of the study showed that heat exchangers which are farm fabricated can be operated with a minimum of maintenance. The average heat transfer coefficient of the unit was 21.3 W/m^2K. The major factors influencing heat transfer were found to be the entrance effects of the air flow and humidity. With an average supply flow rate of 565 L/s, the unit was expected to reclaim 94 GJ of energy per year. The payback on reclaimed energy can make such a system economically viable for many operations presently in Alberta.

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LIST OF SYMBOLS

Symbol	Description	Units
A	area	m^2
B	building shell losses	W/K
C	capacity rate, $M \cdot Cp$	W/K
Cp	specific heat (constant pressure)	$J/kg \cdot K$
Cs	sensible capacity rate	W/K
D	diameter	m
DD	degree-day	day°C
e	effectiveness	
E	energy	J
h	specific enthalpy	kJ/kg
HP	heat production	W
LAT	latent heat	W
LMTD	log mean temperature difference	K
M	mass flow rate	kg/s
Nr	Reynolds number	
NTU	number of thermal units	
q	heat flow rate	W
Q	volume flow rate	m^3/s
R	recovered heat	J
Rh	relative humidity	
SUPP	supplemental heat	W
T	temperature	K
t	time	days
TRR	temperature recovery ratio	
U	overall heat transfer coefficient	$W/m^2 \cdot K$

v	velocity	m/s

Subscript	Description
c	cold fluid
def	condition at deficit temperature
i	inlet condition
in	inside condition
o	outlet condition
out	outside condition
min	minimum
max	maximum
w	warm fluid
x,y	position indicators

1. INTRODUCTION

1.1 Ventilation

Since the practice of feeding livestock first began, farmers in the more temperate climates would have noticed increases in feed requirements during periods of cold. By artificially creating a less harsh micro-environment by means of barns, operators were able to increase the comfort of themselves and the animals, increase the health and productivity of the animals, and decrease feed costs. In recent years, the concept of controlled environments has led to the development of large animal confinement housing units, containing thousands of animals and tended by a single operator.

The close confinement of many animals under one roof has created problems in the quality of air the animals breathe, with the major concern being the levels of humidity, dust, ammonia (NH_3), hydrogen sulphide (H_2S), methane (CH_4), and carbon dioxide (CO_2), as well as the presence of odours and microbes. Although definite and reliable standards for the maximum acceptable levels of the aforementioned substances are not agreed upon at present, large animal housing units are equipped with means for reducing the levels of these substances by continuously replacing the stale barn air with outside fresh air. This can be accomplished either mechanically by fans or "naturally" using the buoyant force of the warmer barn air and wind.

In the case of some animal units, particularly those for poultry and swine, when the outside temperature is low, the ventilation rates can be great enough to create the situation where the heat removed by replacing the warm barn air with cold outside air is greater than that produced by the animals themselves. This heating deficit causes the barn environment to be colder than that for optimum animal performance. In order to maintain the desired barn temperature, supplemental heat can be added from an external energy source such as a furnace. Depending upon the barn and operation, as much as 90% of the heat lost can be the result of the ventilation (Feddes et al., 1984). If present ventilation rates are found to be inadequate for health and comfort reasons, then the amount of heat removed by the ventilation system would be even higher. The possibility of reclaiming this heat has become an area of interest, and will draw more attention if the cost of fuel increases significantly.

1.2 Heat Recovery

The heat from warm, exhausted barn air can be partially recovered and used to heat the cold supply air as it enters the system. This can be achieved using only the temperature differential as the driving force, or the heat transfer can be enhanced with the addition of external energy. Devices which utilize the former are heat exchangers, and those which utilize the latter are heat pumps. Heat exchangers can be very simple devices, requiring only a physical barrier between the two air streams. Heat travels from the warmer to the cooler air mass. Heat pumps are technically complex and expensive devices, although they have the advantage of having the potential to reclaim a greater portion of the energy than a heat exchanger.

With the present energy costs in Alberta, the electricity cost for running a heat pump would, at best, equal the savings from unused natural gas. The high capital outlay for a heat pump could not be justified, making the device an uneconomical alternative. Therefore, the majority of research in agricultural ventilation heat recovery has been devoted to the heat exchanger. The focus of this study is on the heat recovery by agricultural heat exchangers.

1.3 Problems with Heat Exchangers

Research into the development of heat exchangers for the reclamation of heat from barn ventilation exhaust has been continuing for the last forty years. The nature of the exhaust from animal confinement units creates severe problems which have hindered the implementation of heat exchangers on a large scale. The basic problems encountered can be summarized as follows:

i) Durability - Barn air is moist and corrosive, which can greatly reduce the life expectancy of devices which would perform adequately in other, less harsh environments. Attempting to reduce material costs can produce a device which is easily damaged and requires frequent repair.

ii) Fouling - Dust from the animals, litter, and feed can plug a compact heat exchanger in a matter of days. Costs associated with filtering and cleaning can increase operating costs far above acceptable levels.

iii) Freezing - In cold weather, the exhaust air can become cooled below the freezing point of water. As the condensate on the heat transfer surface freezes, the expansion can produce stresses greater than the heat exchanger is able to resist or the ice can simply build up to the point of shutting off all air flow. In addition, when the barn air is exhausted to the atmosphere, any condensate present can fall to the ground and freeze, producing an ice build-up beneath the exhaust outlet which could require frequent removal.

iv) Economics - Energy costs in Alberta are such that heat exchangers that may be marginally viable in other regions can often have unacceptable returns if applied locally. The low energy cost of natural gas (approximately $4/GJ) limits the amount of additional fan power that can be used by the device before heat recovery becomes uneconomical.

1.4 Objectives

The process most often involved in the development of agricultural heat exchangers appears to have been trial-and-error. Even the earliest experiments involved a variety of heat exchanger geometries and flow directions with no conclusions reached as to which was more appropriate for any particular application (Giese and Downing, 1950; Giese and Ibrahim, 1950; Giese and Bond, 1952). The large number of possible combinations of flow pattern, geometry, size, construction materials, flow rates, and even barn environment has made it difficult to attribute the relative success or failure of a test to any particular feature of the design. For this reason, there have been a large variety of heat exchangers tested using this trial-and-error technique, and the failure of a particular design has often resulted in its abandonment and the adoption of a largely arbitrarily selected new design. The objective of this study was to systematically develop a design for a heat exchanger which eliminates the problems outlined above.

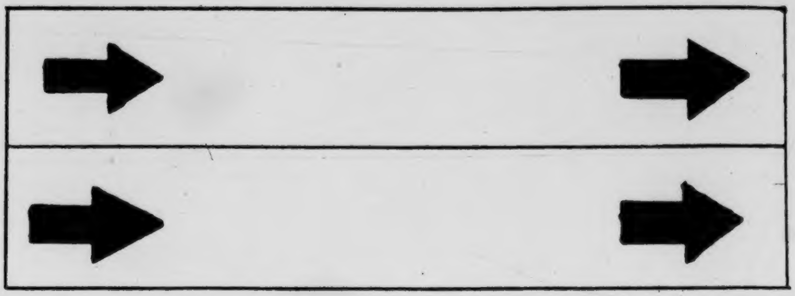
The approach used and the steps taken to fulfill the objective is the topic of this thesis. The presentation of the material is basically in the chronological order of the design process used. The first step was the determination of the possible design types, factors influencing performance, and methods of describing performance. The next step was to review the previous work done on heat exchangers with the focus being the determination of which features of a particular design are desirable and should be considered for the prototype. The information gathered by studying the previous work was then used to develop several computer models to determine the optimum design specifications, leading to the design of the prototype. The next logical steps were the fabrication and testing of the prototype, followed by the presentation and discussion of the results.

2. PREVIOUS WORK

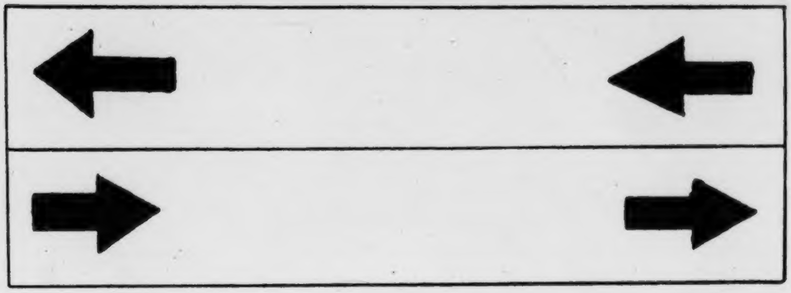
2.1 Types of Heat Exchangers

Heat exchangers are devices in which two fluid streams are physically separated by a highly conductive heat transfer medium. Temperature difference between the two streams causes heat to flow from the fluid of higher temperature to that of lower temperature. Heat exchangers can be described in terms of the relative directions of flow of the two fluid streams and also in terms of geometry and nature of the heat transfer medium.

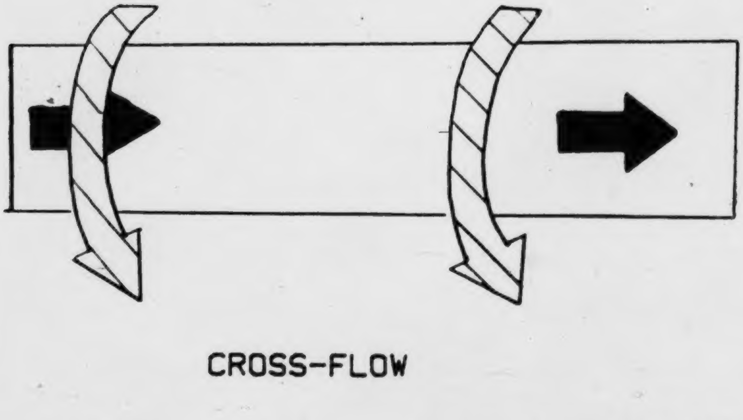
The three basic possibilities for flow direction are parallel-flow (also called concurrent-flow), counter-flow, and cross-flow, as shown schematically in Figure 2.1. Parallel-flow heat exchangers have a high initial temperature difference near the inlets, which creates a high heat transfer rate in that region. The temperatures of the two fluid masses converge towards an intermediate temperature near the outlets, creating a low heat transfer rate in that region. Counter-flow designs have a more uniform temperature difference along their entire length and, therefore, a more uniform local heat transfer rate. Cross-flow designs have non-uniform cross-sectional temperature profiles and can have local charateristics of both parallel- and counter-flow. The thermal performance of cross-flow designs would be bounded by those of parallel- and counter-flow designs having similar flow parameters.



PARALLEL-FLOW



COUNTER-FLOW



CROSS-FLOW

Figure 2.1 Types of heat exchangers based on flow direction

For the purpose of this study, heat exchangers were grouped into five general categories according to the geometry of the heat transfer medium. These five categories are briefly described below:

i) Flat-plate. Flat-plate heat exchangers are those for which the two fluid masses are separated by a thin, flat membrane or plate, through which the heat is directly transferred. Multiple-plate designs have a series of plates which provide narrow channels for the fluids. By alternating the channels for the warm and cold fluids, heat transfer is made to occur on both sides of the fluid boundary. Flat-plate designs can be either parallel-, counter-, or cross-flow.

ii) Regenerative. Regenerative heat exchangers employ a heat sink which is alternately placed in each of the fluid streams. Industrial regenerative heat exchangers typically consist of a wheel of metal mesh which rotates such that one half of the wheel is in each fluid stream. The heat absorbed by the portion in the warmer stream is released as the rotation of the wheel causes contact with the cooler stream.

iii) Shell-and-tube. Shell-and-tube heat exchangers consist of a bank of tubes through which one of the fluids flows. Heat is transferred through the walls of the tubes to the other fluid flowing around the tube bank, usually in a cross-flow pattern to increase the heat transfer achieved.

iv) Air-liquid-air. The term air-liquid-air is used here to describe those designs in which the heat is transfered from the warmer source, through a solid barrier, to a liquid. The liquid is moved to the vicinity of the cooler stream and the heat is transfered via another solid barrier. The liquid can be mechanically pumped or, as in the case of thermosiphons, it can be circulated using the buoyant forces produced by temperature change. Thermosiphons (a type of heat pipe) are sealed metal tubes (often finned) containing a specific charge of freon. The tubes are inclined with the lower portion in the warm stream and the upper portion in the colder stream. The temperature differential causes a slight difference in the the local vapor pressures of the freon for the two regions. The liquid freon in the lower portion boils, the vapor produced condenses on the cooler walls of the upper region and returns to the bottom by running down the inside of the tube. This continual change of phase creates high heat transfer coefficients in both regions, resulting in minimal thermal resistance of the system due to the transfer medium.

v) Earth-tube. Earth-tube heat exchangers are used basically only for ventilation purposes. The supply air for the structure is drawn through long tubes buried typically at a depth of 3 m. The term 'heat exchanger' is somewhat inaccurate for these designs since, unlike any of the four previously discussed types, no heat is recovered from the ventilation exhaust. By having the supply air drawn through the earth, the sharp variations in temperature over the year are tempered by the large heat storage capacity of the system.

2.2 Heat Exchanger Performance

2.2.1 Heat Transfer

The rate of heat exchange between the two air streams is dependent upon the thermal resistance of the physical barrier seperating the streams and that of the solid-gas interfaces on each side of the barrier. In an agricultural heat exchanger, the thermal resistance of the barrier itself is normally negligible when compared to that produced by the solid-gas interfaces. For simplification, the effects of the individual components of the thermal resistance are combined and represented by the overall heat transfer coefficient (U). The method for predicting heat exchanger performance, acceptable for most industrial applications, is outlined in ASHRAE (1981). For an incremental length of heat exchanger, the local heat transfer can be calculated by the following equation:

$$dq = U * (T1 - T2) * dA \qquad 2.1$$

where:

dq = incremental heat transfer rate (W),

dA = incremental heat transfer area (m^2), U = overall heat transfer coefficient ($W/m^2 \cdot K$),

T1 = local temperature of exhaust air stream (K), and

T2 = local temperature of supply air stream (K).

For most applications, the mode of heat transfer is forced convection between both operating fluids and the heat transfer surface, so that the overall heat transfer coefficient is relatively constant along the heat exchanger and can, therefore, be approximated by an average for the entire heat transfer surface. Also, the specific heat capacities of these two operating fluids usually can be assumed to be constant. If the above two assumptions are valid, then equation 2.1 can be integrated to obtain the following:

$$q = U * A * LMTD \qquad 2.2$$

where:

q = total heat transferred (W),

U = heat transfer coefficient ($W/m^2 \cdot K$),

A = total heat transfer area (m^2),

LMTD = log mean temperature difference (K),

and:

$$LMTD = (T_x - T_y)/ \ln (T_x/T_y) \qquad 2.3$$

where Tx and Ty are the temperature differences between the hot and cold fluids at each end of the heat exchanger.

The use of equation 2.2 can be questionable when applied to agricultural heat exchangers due to the possible lack of validity of the two assumptions made in its derivation. Firstly, the heat transfer coefficient between the two air streams increases once the exhaust air is cooled below its dew point. The resulting condensation on the heat transfer surface has the effect of increasing the rate of heat transfer. Secondly, possible error can result from the apparent increase in the specific heat of the exhaust air stream as the latent heat is released during condensation. Cooling one kilogram of unsaturated air from 20°C to 19°C releases approximately 1 kJ of energy, whereas to cool the same mass when saturated releases 3.3 kJ.

If the assumption is made that equation 2.2 can be used for agricultural heat exchangers with acceptable error, the calculation for heat transfer area needed for a given application requires only an estimation of the value for the heat transfer coefficient. The uncommon conditions which exist in animal environments as compared to other environments and the wide variations in heat exchanger tests under those conditions complicate even a rough prediction for the heat transfer coefficient. Heat exchangers used industrially typically have homogeneous fluids and, although phase change is encountered, it usually occurs isothermally. Rigorous calculations for predicting heat exchanger performance for barn ventilation are made difficult by inclusion of the effect of the condensation of the moist barn air as it is cooled, while the presence of varying thicknesses of dust on the heat transfer surface further increases the uncertainty of conventional methods of analysis.

By neglecting the effect of dust and humidity, standard heat transfer equations (Ozisik, 1977) can be applied to a simple air-to-air heat exchanger to predict the heat transfer coefficient. Using data supplied from a test on a shell-and-tube heat exchanger (McGinnis et al., 1981), the equations in Ozisik (1977) yield an expected heat transfer coefficient of 7 $W/m^2 \cdot K$. When the exchanger was tested in a hog barn, the performance data, as presented by McGinnis et al., suggested that the overall heat transfer coefficient was approximately 18 $W/m^2 \cdot K$. In a laboratory test of a flat-plate design, Giese and Bond (1952) measured heat transfer coefficients for both dry and humid exhaust air. The values for the humid air ranged from 10 to 14 $W/m^2 \cdot K$ and were approximately 25% higher than for the dry air tests. Another design, very similar to the one used by Giese and Bond (1952), also was tested under laboratory conditions (Ogilvie, 1967) and, although no mention was made of humidity levels, heat transfer coefficients from 22 to 29 $W/m^2 \cdot K$ were measured. For that test, the suggestion was made that induced turbulence produced by the fans could have accounted for the higher values. A single flat-plate design was tested in a broiler barn (Hodgkinson and Small, 1984) and the measured heat transfer coefficients ranged from 12 to 21 $W/m^2 \cdot K$ as the average air velocity through the exchanger was varied from 1.5 to 3.5 m/s. The average relative humidity for these tests was 80%.

The effect of fouling on the heat transfer coefficient appears to be much less than that produced by humidity and tends to be insignificant in comparison. Stauffer and Vaughan (1980) reported very little difference in heat exchanger performance with fouling, and although the presence of the dust reduced the flow area and increased fan power considerably, the resulting increased velocity appeared to keep the heat transfer almost constant.

2.2.2 Describing Heat Exchanger Performance

Direct comparison of different heat exchanger tests is complicated by the lack of a standard method for describing the performance in terms of the amount of heat transferred. Equation 2.2 is useful for cases where the inlet temperatures are known and constant, but a better method would use dimensionless parameters and be universally accepted. ASHRAE (1981) describes the amount of heat transferred by the 'exchanger heat transfer effectiveness', e, where e is calculated according to the following equations:

$$e = \frac{C_w * (T_{wi} - T_{wo})}{C_c * (T_{co} - T_{ci})} \qquad \text{when } C_c = C_{min} \qquad 2.4$$

$$e = \frac{C_c * (T_{co} - T_{ci})}{C_w * (T_{wi} - T_{ci})} \qquad \text{when } C_w = C_{min} \qquad 2.5$$

where:

C = fluid capacity rate, M·Cp (W/K),
T = temperature,
w = hot fluid,
c = cold fluid,
min = minimum
i = inlet condition, and
o = outlet condition.

Another dimensionless parameter that is used to describe heat exchangers is the 'number of heat transfer units', or NTU (ASHRAE, 1981). The NTU is calculated according to equation 2.6:

$$NTU = \frac{A * U}{C_{min}} \qquad 2.6$$

where A is the total heat transfer area (m^2).

If the assumptions used to satisfy equation 2.2 are valid, the effectiveness of a heat exchanger becomes simply a function of capacity rates of the two fluids, the flow directions, and the NTU. For such a case, the effectiveness can be easily predicted for a given application provided the heat transfer coefficient is known. For agricultural heat exchangers, equations 2.4 and 2.5 are not readily applicable since the apparent heat capacity of the exhaust air depends upon the humidity and degree of cooling. If the assumption is made that no heat is transfered to the surroundings from the heat exchanger (i.e. thermally insulated), equations 2.4 and 2.5 can be simplified as follows:

$$e_1 = \frac{T_{co} - T_{ci}}{T_{wi} - T_{ci}} \qquad \text{when } C_c = C_{min} \qquad 2.7$$

$$e_1 = \frac{T_{wi} - T_{wo}}{T_{wi} - T_{ci}} \qquad \text{when } C_w = C_{min} \qquad 2.8$$

where:

e_1 = effectiveness as per this definition.

Since Cmin usually refers to the cold supply air in agricultural heat exchangers, equation 2.7 would be the applicable definition and is often used when describing heat exchanger performance (Ogilvie, 1967; Lampman, 1978; Witz et al., 1974). By describing performance in this fashion, the amount of heat being transferred by the exchanger is stated as the fraction of the heat which would be transferred by an infinitely long counter-flow exchanger having the same flow rates. This method has limited practical usage since a heat exchanger with a high ratio of exhaust to supply flow rate would have an effectiveness approaching 1.0 and yet very little heat would actually be reclaimed.

Another ratio used, also called effectiveness, involves enthalpies in the calculation, rather than just temperature (McGinnis et al., 1981; Hodgkinson and Small, 1984). The equation to calculate effectiveness according to this definition is given by equation 2.9.

$$e_2 = \frac{h_{co} - h_{ci}}{h_{hi} - h_{co}} \qquad 2.9$$

where:

e_2 = effectiveness as per this definition, and

h = specific enthalpy (kJ/kg·K).

Equation 2.9 also has limited practical usage since, again, the true amount of heat recovered is not reflected accurately and, also, the humidity level of the barn air would greatly affect its enthalpy and, therefore, the calculated effectiveness. An infinitely long counter-flow heat exchanger could have an effectiveness as low as 0.4 in a moist barn and as high as 1.0 in a dry barn.

2.2.3 Temperature Recovery Ratio, TRR

The method used to describe the performance of a heat exchanger should provide an easy and quick method for determining the heat recovered by the unit. In a barn, the concept of conservation of mass requires the total mass of dry air leaving the structure to equal the total mass of dry air entering. In the case of a barn equipped with a heat exchanger, if the exhaust flow rate is greater than the supply flow rate, the difference must be made up by infiltration through the walls and cracks or the supply inlets. The net result of this process would be to to lower the effective temperature rise of the tempered supply air by the ratio of supply to exhaust flow rates. If the supply flow rate is greater than the exhaust flow rate, the supply air temperaure rise would reflect the true amount of heat being recovered by the exchanger.

In this thesis, the proposed method of describing heat exchanger performance has been termed the 'temperature recovery ratio' (TRR) and is defined as follows:

$$TRR = \frac{M_c * (T_{co} - T_{ci})}{M_{max} * (T_{wi} - T_{ci})} \qquad 2.10$$

Many of the problems resulting from describing the perfomance of an agricultural heat exchanger using the effectiveness of the unit are eliminated with this definition of the TRR. The term 'temperature recovery ratio' was chosen over 'effectiveness' in order to clarify which method was used to calculate the stated performance, and it is the former term which will be used for the remainder of this thesis. The TRR of an infinitely long counter-flow heat exchanger would approach 1.0 only when the two mass flow rates are close to equal. Most heat exchangers would have a TRR in the range of 0.2 to 0.7 and the determination of desirable values will be dealt with in section 4.2.

2.3 Heat Exchanger Tests

2.3.1 Flat-plate Designs

The majority of commercially produced agricultural heat exchangers are of the flat-plate type, mainly because of the high surface area obtainable in a compact volume. By placing the series of plates at close spacing (less than 10mm), the problem is encountered whereby almost total plugging with dust of the exhaust side occurs in a matter of weeks. A typical example is the design developed by Guertin et al. (1980) which was built to handle air flows of 15 m^3/s. Once installed in the hog building for which it was developed, the 4mm spacings proved unsuited for the environment. Filters were added, which increased the fan electrical consumption but lessened the frequency of thorough washings which, without filters, were necessary every few weeks.

Several multiple plate heat exchangers have been built and tested which utilize various forms of readily available aluminum sheeting to provide the heat transfer surface. After a few years, the corrosiveness of the animal environment has become apparent. Witz et al. (1974) briefly describe the utilization of such an aluminum heat exchanger in the relatively clean environment of a dairy barn. The life of the aluminum for that and similar applications was given as not more than three to four years. Smith (1983) reported the use of an aluminum core heat exchanger in a hatchery operation. Under those test conditions, the problems of corrosion and fouling were not recorded. Combining the high exhaust temperature of 33.5°C (65%RH) with the mild Atlanta, Georgia climate, the problem of ice formation was not applicable and Smith was able to use a supply rate of 2.0 m^3/s and an exhaust rate of 1.0 m^3/s with the intent of optimizing the latent heat recovery. Smith reported 61% of the heat recovered was, in fact, latent. This implies a larger increase in supply temperature than decrease in the exhaust temperature, even with the unequal flow rates. Even at the modest cost of U.S. $3200 ($1.60 per L/s) the payback period of the system was 6 years using natural gas as the energy source. The recommendation was made to examine the possibility of using plastics rather than aluminum to decrease the initial cost of the unit.

Flat-plate designs which do use plastic and do not have problems with fouling are those which use thin polyethylene film as the heat transfer medium. These types require low air velocities (less than 3m/s) to prevent fluttering of the sheeting. This requires large flow areas, thus eliminating the problem of plugging with dust, but also leads to very bulky devices and high costs of supporting materials.

A field analysis of a polyethylene film heat exchanger was performed in Manitoba (Wilson 1982). The design was one developed by Saskatchewan Agriculture and was commonly known in Western Canada as the plan S-981, publication of which has been discontinued recently. The major problems encountered were low durability, complicated design, and high cost. The site of Wilson's test was a 144 sow farrowing operation. The heat exchanger was constructed in a well equipped facility and yet required an estimated 250 to 300 man-hours of labor to build. In addition, the material costs were listed as $2530. The device was mounted on a concrete platform outside the barn and drew supply air from the attic at a rate of $0.75 m^3/s$ and had an exhaust rate of $0.9 m^3/s$. With a sheet spacing of 12.5mm, no dust problems were encountered, although a build up was expected to occur in the exhaust duct after a few years, which would require removal. Freezing of the condensate was observed at temperatures below -30°C although no structural damage was recorded. In one year of operation, the unit was estimated as having saved 160 GJ of energy, which would give a payback period of 8 years, at an energy cost of $4 per GJ.

A more recent study was performed by Hodgkinson and Small (1984) on a design consisting of two channels separated by a single sheet of polyethylene. The test was conducted in a broiler barn. Polyethylene sheeting was chosen for its low adherence to dust and ice. Also, the condensation was expected to wash the dirt off and help keep the plastic clean. The gap between the plastic sheeting and the plywood duct wall was between 75 and 100mm wide and 600mm high. Detrimental dust build-up along the bottom was not expected as the exhaust side was provided with drainage. Also, as is the practice with broiler operations, the barn was left empty for two to three weeks between batches which allowed time for any dust on the sheeting to dry and be blown out of the system. The system operated for two years without cleaning, in which time 100 mm of dust had, however, accumulated along the bottom and required four man hours to remove. The unit was 12.2 m long and could easily be disassembled for maintenance and cleaning. The material costs were given as $250 and the labor required for construction was listed as 20 man-hours. The durability of the design must be questioned, however, as a problem was reported with the plastic sheeting dislodging and partially blocking the flow. Without the regular testing of flow rates during the test period, the problem would not have been noticed. The expected life of such a system would not be great since disassembly would be required for regular maintenance and inspection. As with the Plan S-981 unit, care would have to be taken to ensure mice did not enter the device and damage the film.

2.3.2 Regenerative Designs

As noted in the introduction, the most common regenerative heat exchangers used in industry are those which utilize a rotating drum of metal mesh. The main reason this type has never been applied to agricultural applications is the problem with dust plugging the mesh and the corrosiveness of the air. The type which has been developed and studied in Western Canada is that which uses a heat sink consisting of two beds of 10 to 50 mm mean diameter rocks placed alternately in the exhaust and supply air streams by means of a swinging damper. Extensive field studies of this type were reported by Lampman (1978,1982). The problems of dust and ice build-up were severe in those heat exchangers studied and, in one instance, the addition of 1100W heating cables was used to thaw one rock bed and prevent further freeze-up. As the air flows reversed direction in response to the changing of the damper, mixing of the exhaust and supply air masses and reintroduction of some dust and foul air into the barn is inevitable. Also, Lampman (1978) noted little effect of moisture content of the exhaust air upon the heat recovery of the units. This observation was supported by laboratory tests of similar rock beds (Moysey et al., 1980).

Another rock bed heat exchanger, installed at the North Dakota State University was described by Witz et al. (1974). Although the temperature recovery ratio was lower than for the units described by Lampman (33% as compared to 40%), severe icing problems again were encountered. Even when actual plugging of the system with ice was not a problem, the condensate dripping from the rock beds would freeze and require removal. Typically, the velocity of the air stream through the rock bed is required to be 0.5 m/s or less to keep the pressure drop low enough to be handled by simple axial flow fans (approximately 100 Pa static). This would result in larger structures than would be necessary using designs capable of handling higher velocities. The partial reintroduction of foul air, dust and moisture; the apparent inability of the device to reclaim latent heat; the potential for high fan power requirements; and the fouling and ice problems are all disadvantages which are not compensated for by any readily-apparent, unique advantages of these designs. Also since none of the sources referred to the initial costs of regenerative heat exchangers, a reasonable assumption would be that this is not a favorable aspect of these designs.

2.3.3 Shell-and-tube Designs

Typical shell-and-tube designs are those in which the exhaust air passes through PVC or comparable plastic pipe, around which flows the supply air, usually in a cross-flow pattern. The pipe diameter is typically 50mm, so dust problems severe enough to plug the pipes are seldom encountered and, should freezing occur, the pipe is strong enough to maintain performance after thawing is complete.

Extensive research with polyethylene tube heat exchangers has been carried out by McGinnis et al. (1981, 1983). As with the flat-plate designs, polyethylene was chosen for its low adherence to dust and ice. The tubes were 1.5 m long, 50 mm in diameter and were placed in a plywood-baffled shell such that the supply air flowed over them in a counter-flow, cross-flow multiple pass pattern. The exhaust air flowed vertically downward.

A suggested design improvement was that the tubes be built in modules of 36 which could be combined to give any desired air flow capacity, although the tests reported were done on 196- and 169-tube prototypes. The temperature recovery ratios of both designs were around 30%, as calculated from the reported data, but ice formation still occurred and required consideration. The high humidity present in the test barns caused the plywood frames to swell and, where the tubes passed through holes in the plywood, the swelling caused them to become slightly constricted. When ice build-up had become sufficiently great to considerably restrict the flow, the ice had to be thawed enough, not only to become dislodged, but also to become small enough to slide past these constrictions. The recovery of latent heat associated with freezing was viewed as beneficial and one of the prototypes was equipped with a pressure transducer so that defrosting (which was achieved by reducing the supply air flow rate) could be initiated once ice build up reached a maximum acceptable level as determined by fan load criteria. The recommendation was given that the bottom of the tubes be flared to minimize the thawing required to dislodge the ice and allow it to fall out via the exhaust route. This approach, although allowing the recovery of the heat of fusion of whatever ice leaves the system, would increase the initial cost of the unit and also produces the problem of regular removal of ice mounds at the outlet, similar to the case of the regenerative types.

McGinnis et al. (1981) also did laboratory tests on a single module of 36 tubes with specific concern paid to the pressure losses associated with the design and optimization of flow. The ratio of fan power to heat recovered from the exhaust air was found to be in the range from 9 to 10, with the pressure loss being 30 times greater on the shell side than through the tubes. The baffles, while increasing the effectiveness, also increase the pressure losses and the recommended cross-flow velocity was given as 10 m/s considering the two effects.

The pressure losses associated with dust formation were negligible. After being used four months in a dry-feed swine environment, an estimated 0.5 mm of dust had accumulated. This represented a 2% reduction in flow area, while the same thickness of dust would represent a 25% reduction in flow area with the flat-plate design of Guertin et al. (1980).

A less compact shell-and-tube heat exchanger was described by Meyer et al. (1983). This design was, in some ways, more similar to the polyethylene film flat-plate design than to the shell-and-tube designs described above. This particular design had the shell side of the exchanger consisting of a 1.5m wide alley, running the 41.4m length of a farrowing-nursery building, through which the exhaust air flowed. Running along the ceiling and one side of the alley were eight, 300mm diameter polyethylene-film tubes which acted as inlet ducts. This design had a cost of $2.50 per

L/s although, as with the flat-plate design of Smith (1983), the practicality of such a device in Alberta climates is questionable. The average January temperature for the site of the test (Pennsylvania) was -7°C and acceptable condensation was experienced for the two month test period, even though the exhaust duct doubled as a service alley with the cold supply air running through the polyethylene-film tubes directly overhead.

2.3.4 Air-liquid-air Designs

Albright and Behler (1984) described an air-liquid-air heat exchanger which used a water-ethylene glycol mixture to transfer heat between the exhaust and supply air flows of a greenhouse. The system utilized surplus refrigeration evaporators placed in the air streams, through which the liquid was circulated. As dust concentrations in greenhouses are lower than for animal confinement environments, fouling of the fins on the evaporator cores was not a problem. Since heat was physically carried by the intermediate fluid, the two air flows did not have to be in close proximity. The finned evaporators provided high surface areas for heat transfer without any special ducting problems. The liquid was circulated by means of a small pump which could be regulated for variable flow rates of the liquid. When the flow rate of this intermediate fluid was lowered beyond a certain point, the temperature recovery ratio of the unit would begin to drop. The reduction of the temperature recovery ratio could be an effective way of thawing or preventing ice build-up in very cold weather without altering the air flow rates.

The air-liquid-air heat exchangers which were applied to animal housing units were mostly of the thermosiphon type. One such thermosiphon unit was tested in the mild climate of Blacksburg, Virginia (Stauffer and Vaughan, 1981). Although the thermosiphons were finned and the application was in a hog barn, the results indicated that the installation and maintenance of filters was not economical. During the tests, accumulation of dust caused the air flow rate through the exhaust side to be reduced to 83% of its clean value but the rates of heat transfer were not affected significantly. A coarse filter was installed, however, on the basis that the reduced life of the fan and cleaning of the blades was considered sufficient to warrant filtering.

A thermosiphon design which was tested in a number of Canadian locations was developed by Larkin et al. (1975, 1979). Unlike the Virginian model, the use of filters was considered necessary for both poultry and swine environments. Vacuuming was not an effective means of cleaning the finned aluminum heat pipes, so a spray wash schedule was implemented. Once installed, the filters also were vacuumed every two days and the entire system was washed every month. Aluminum mesh filters were found to be not durable enough, so cheaper domestic furnace filters were installed and had to be replaced weekly in addition to the regular vacuuming. Freezing was another problem in the early tests done in Ottawa, so a thermostat was installed to regulate the supply air flow. Although aluminum was the metal used for both the fins and the pipes, no significant corrosion was experienced after 3 years, perhaps due to the frequent cleanings.

As noted earlier, the transfer fluid in air-liquid-air heat exchangers has a much higher heat transfer coefficient than the air films (especially when phase change of the transfer fluid is involved), so that the resulting overall thermal resistance can be approximated by the air film values only. A certain amount of performance is lost by the fin efficiencies being less than 100%, so that the actual heat transfer for the finned area of a thermosiphon is less than that for an equal surface area of flat plates. Added to that disadvantage is the relative complexity of charging the pipes with freon and sealing them, which requires specialized equipment and skilled labor. Interest in thermosiphon heat exchangers appears to be decreasing in favor of other designs.

2.3.5 Earth Tube Designs

Earth tube heat exchangers are those which draw the supply air through long pipes buried in the ground at a depth of around 3 m. The term heat exchanger is somewhat inaccurate in this context since, unlike any of the four previously discussed types, no attempt is made to recover any of the heat from the ventilation exhaust. By having the supply air pass through the earth, the sharp variations in temperature over the year are averaged out by the large heat storage capacity of the system.

Goetsch and Muehling (1984) described several earth tube systems in use in Illinois. The advantage of these devices over conventional heat exchangers is that, not only are winter heating bills reduced but, also, the hot summer air is tempered to provide a cooler environment. This is particularly usefull for regions further south than Alberta, where ambient conditions are often in excess of 30°C. Performance data provided for one of the Illinois installations indicated that the heating provided by the earth tube device accounted for approximately one half of the total heat load required for the system. By the end of the heating period, however, the earth tubes were only providing 12% of of the total heat required to bring the incoming air to barn temperature.

The weather data provided by ASHRAE (1981) indicate that Illinois experiences 3300 degree days, while the Edmonton area experiences 5600 degree days (based on a base temperature of 18°C). The length of time for which an earth tube system could provide significant heat during the long northern winter would have to be questioned, whereas a true heat exchanger would provide approximately 40% of the heat to the supply air, even in the March situation described above. The tubes would have to be buried deeper in colder climates which would increase the installation cost. In addition, the requirement for cooler air is not sufficiently common in the northern climate to justify the power required to draw the air through the long lengths of tubing.

2.3.6 Summary

The possible design types for agricultural heat exchangers, as outlined earlier, are the flat-plate, shell-and-tube, regenerative, air-liquid-air, and the earth-tube designs. The earth-tube designs are more suitable for warm regions. The air-liquid-air designs are difficult to adapt to animal environments and have few advantages. The regenerative designs have unique disadvantages which are not compensated for by unique advantages. The shell-and-tube and flat-plate designs appear to be the most promising for the development of a viable agricultural heat exchanger.

3. COMPUTER MODELS

3.1 Flow Direction Analysis

As mentioned previously, three possible relative directions for air flows in a heat exchanger are parallel-flow, counter-flow, and cross-flow. Of these, the optimum flow direction for a given set of conditions could be defined as that which would require the least amount of heat transfer area, thereby having the lowest initial cost (all other parameters being equal). Although agricultural heat exchangers from each type have been tested, the determination of the optimum flow direction has not been carried out previously.

3.1.1 Model Development

Since cross-flow heat exchangers have non-uniform temperature profiles, the measured temperature changes which occur for both the exhaust and supply sides are the result of the averaging of the temperature of the air as it leaves the device. Therefore freezing could occur in a cross-flow heat exchanger even though the effective temperature of the exhaust is still above the freezing point. The complexity of the temperature variations for cross-flow would make modelling difficult and, since the apparent temperature recovery ratio of a cross-flow heat exchanger would be bounded by those with parallel- and counter-flow of equal heat transfer area, only the latter two were considered for determining the optimum.

The applicability of standard equations to agricultural heat exchangers was questioned in the introduction because of the effects of the moisture in the exhaust air. For that reason, computer models were developed to simulate the heat transfer processes for the parallel- and counter-flow situations. The results of these simulations were then used to determine which pattern was better for a given application and also to estimate the error involved in using the standard equations.

The geometry selected for the simulation model was that of a single plate, such as that described by Hodgkinson and Small (1984). The assumptions used in the model were as follows:

i) The density of the air along one side of the exchanger was assumed to be constant. This allows for the assumption of constant mean velocity.

ii) The cross-sections of the flow channels were selected to be rectangular, with the heat transfer occurring along the common side. The height of the common side was selected as 0.7 m.

iii) The velocity of the air was selected to be a constant 4.0 m/s, with a uniform temperature profile for a cross-section.

iv) Once formed, the effect of the condensate was neglected. The condensate was assumed to drain out at the location along the exchanger where it was formed.

No attempt was made to model the effect of condensation on the local heat transfer coefficient for the reason that the great uncertainty in predicting overall heat transfer coefficients would make the selection of local heat transfer coefficients very arbitrary. A constant heat transfer coefficient of 25 $W/m^2 \cdot K$ was used for the entire length of the heat transfer surface.

The computer programs (Appendix A) were written in the Continuous Simulation Modelling Program (CSMP) language. CSMP was used mainly for the ease with which the psychrometric data could be handled (Lambert 1975). The input parameters which were varied were the barn relative humidity, the barn temperature, and the outside temperature. For the parallel-flow case, these inputs were taken as the inlet conditions to the heat exchanger and the heat transfer for the first incremental length was determined from equation 2.1. The temperatures of the two air masses then were adjusted according to the following:

$$DT = Dq / [Cp * M] \qquad 3.1$$

where:

DT = temperature change (K),

Cp = local apparent specific heat ($J/kg \cdot K$),

Dq = heat transferred (W), and

M = mass flow rate (kg/s).

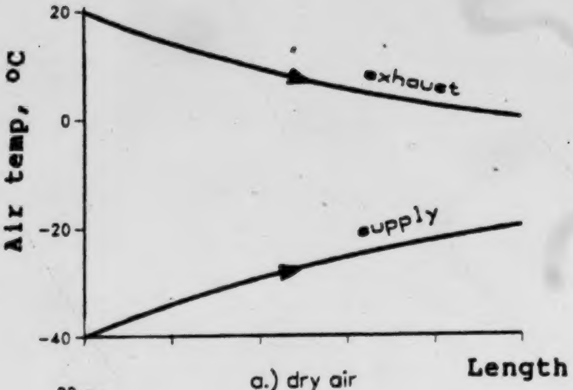
The apparent specific heat capacity of the air mass was determined from tabulated psychrometric data and empirical equations (ASHRAE 1981). The modified temperatures were then used as the inlet conditions for the next incremental length and the process was continued until termination.

For the counter-flow model, the desired outlet condition of the exhaust stream was preselected (usually chosen to be 0°C) to avoid the use of an iterative technique to solve for it and the heat exchanger length. The outlet temperature of the supply air for the first increment could then be determined from basic thermodynamics. Once the starting conditions for the first increment were determined, the process was continued as for the parallel case with the exception that the supply air was followed in the upstream direction.

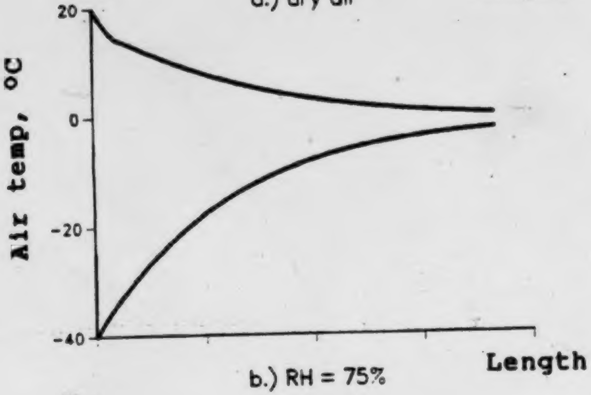
The desired temperature recovery ratio for the purpose of this model was considered to be that which would recover the greatest amount of heat without cooling the exhaust air below the freezing point.

3.1.2 Results

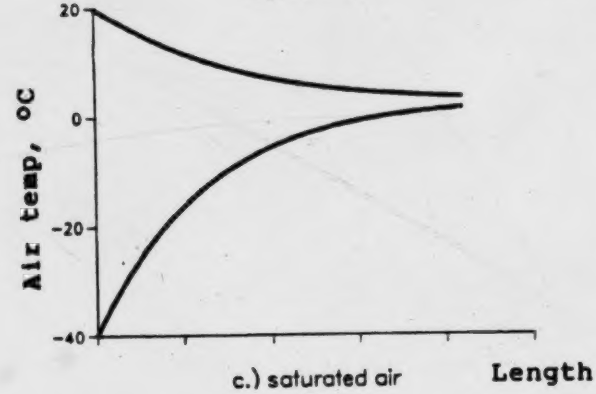
Examples of the temperature profiles obtained from the parallel-flow model are given in Figure 3.1, and those for the counter-flow are given in Figure 3.2. These particular graphs are for a barn temperature of 20°C, outside temperature of -40°C, and relative humidities of 0, 75, and 100%. Figures 3.3 and 3.4 show the discrepancies in heat transfer area required as determined by the simulation models and by using equation 1.2. The close agreement in the



a.) dry air

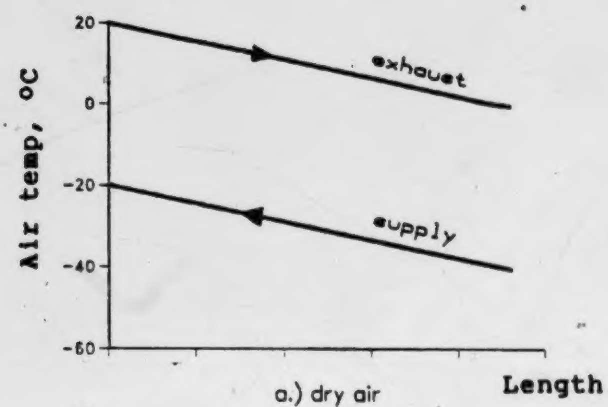


b.) RH = 75%

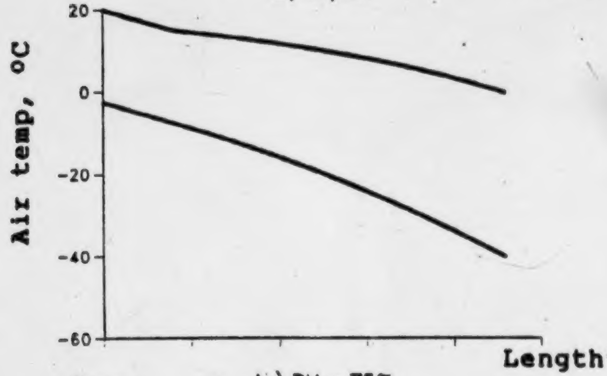


c.) saturated air

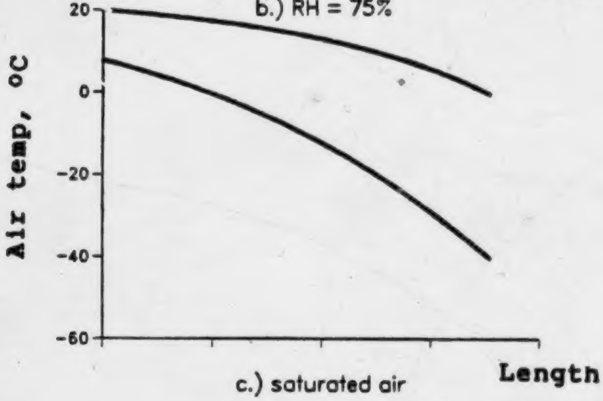
Figure 3.1 Simulated temperature profiles for a parallel-flow heat exchanger



a.) dry air



b.) RH = 75%



c.) saturated air

Figure 3.2 Simulated temperature profiles for a counter-flow heat exchanger

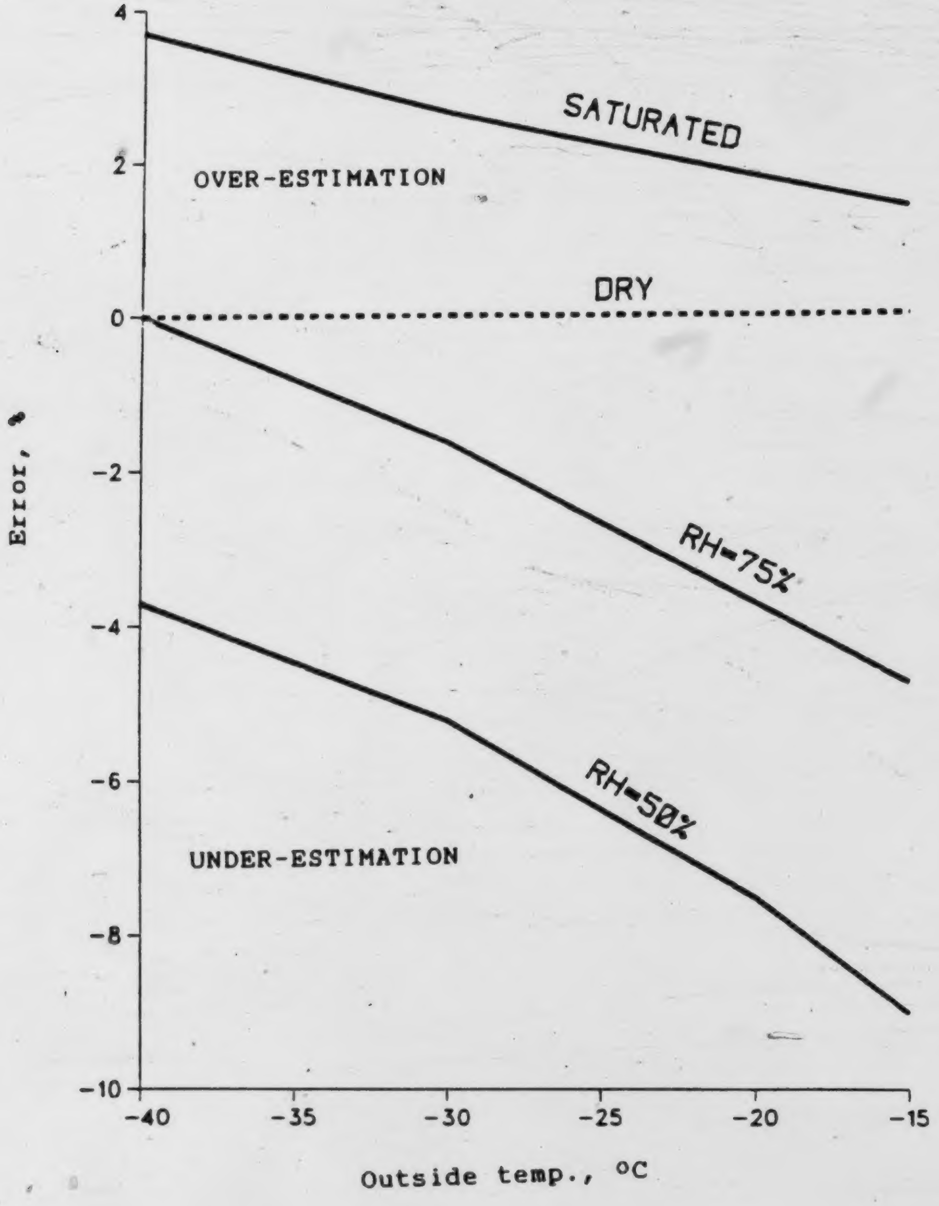


Figure 3.3 Error in predicted heat transfer area for LMTD equation for a parallel-flow heat exchanger.

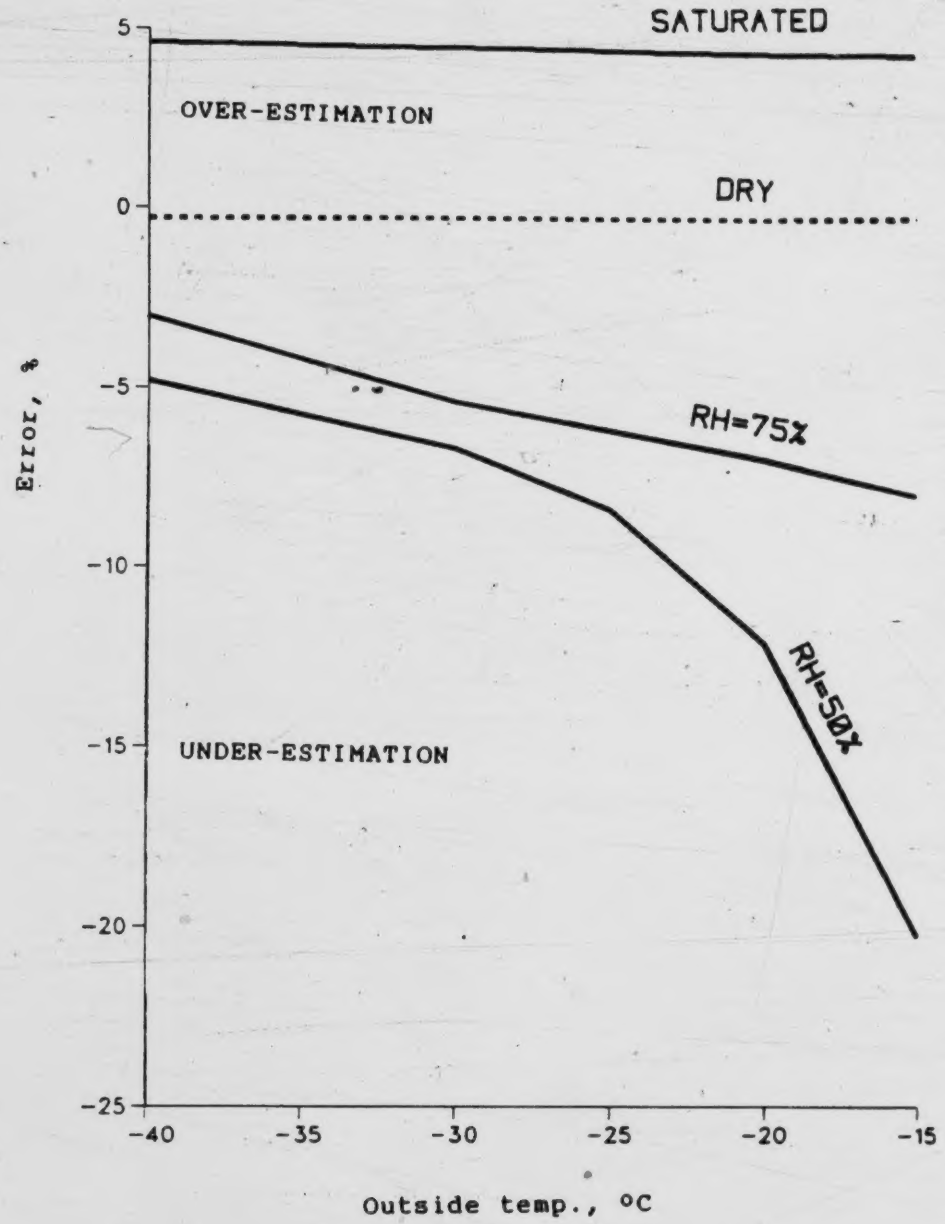


Figure 3.4 Error in predicted heat transfer area for LMTD equation for a counter-flow heat exchanger

case of the dry air for both flow directions is an indication of the correctness of the computer models, since with dry air the assumption of constant specific heat capacity is valid for both methods and each should yield the same result.

For saturated barn air, as the exhaust air cools, the exponential decrease in the humidity ratio results in a decrease in the apparent specific heat of the exhaust air. This creates a larger temperature difference between the two air streams than that assumed by equation 2.2, thereby resulting in an overestimation of heat exchanger size when using that equation. The sudden increase in apparent specific heat capacity of the exhaust air at the onset of condensation results in an effective temperature difference less than that assumed by equation 1.2, resulting in an underestimation of heat exchanger size. As can be seen in Figure 3.3, under certain conditions the two effects cancel each other and the resulting error approaches zero.

The direct application of the quantitative results shown in Figures 3.3 and 3.4 to other design situations is limited. The variation in error with outdoor temperature as could be inferred from the graphs results, at least in part, from the variation in heat transfer area required to satisfy the requirement for optimum heat recovery outlined above. The results as presented should be considered as an indication of the general magnitude of the error that can be expected from using equation 1.2. As can be seen, the errors were generally greater for the counter-flow case, indicating a greater effect of latent heat on temperature differential for that type.

The desired temperature recovery ratio, defined as that which recovers the most heat without freezing the exhaust, is presented in Figure 3.5. In a barn, a heat exchanger which had an operating temperature recovery ratio higher than that indicated in this diagram would be undesirable. Figure 3.6 shows the region for which a parallel-flow heat exchanger would not be capable of achieving this optimum temperature recovery ratio. For such regions the higher potential for heat recovery of the counter-flow pattern may be advantageous. As can be seen, the more humidity in the exhaust, the greater the potential for heat recovery before the freezing point is reached and hence the downward shift in the range in which parallel-flow could achieve the required temperature recovery ratio. As well, the curves in Figure 3.6 are for equal supply and exhaust flow rates. An increase in the ratio of exhaust to supply flow rates would have a similar effect to that of increasing the humidity.

A comparison between the size of counter- and parallel-flow heat exchanger required to achieve the maximum desired TRR for 75% barn humidity is represented in Figure 3.7. The shaded area represents that region for which the heat transfer areas predicted by the simulation models were within 10% of each other. The unshaded area in the upper right is where counter-flow designs require a smaller heat

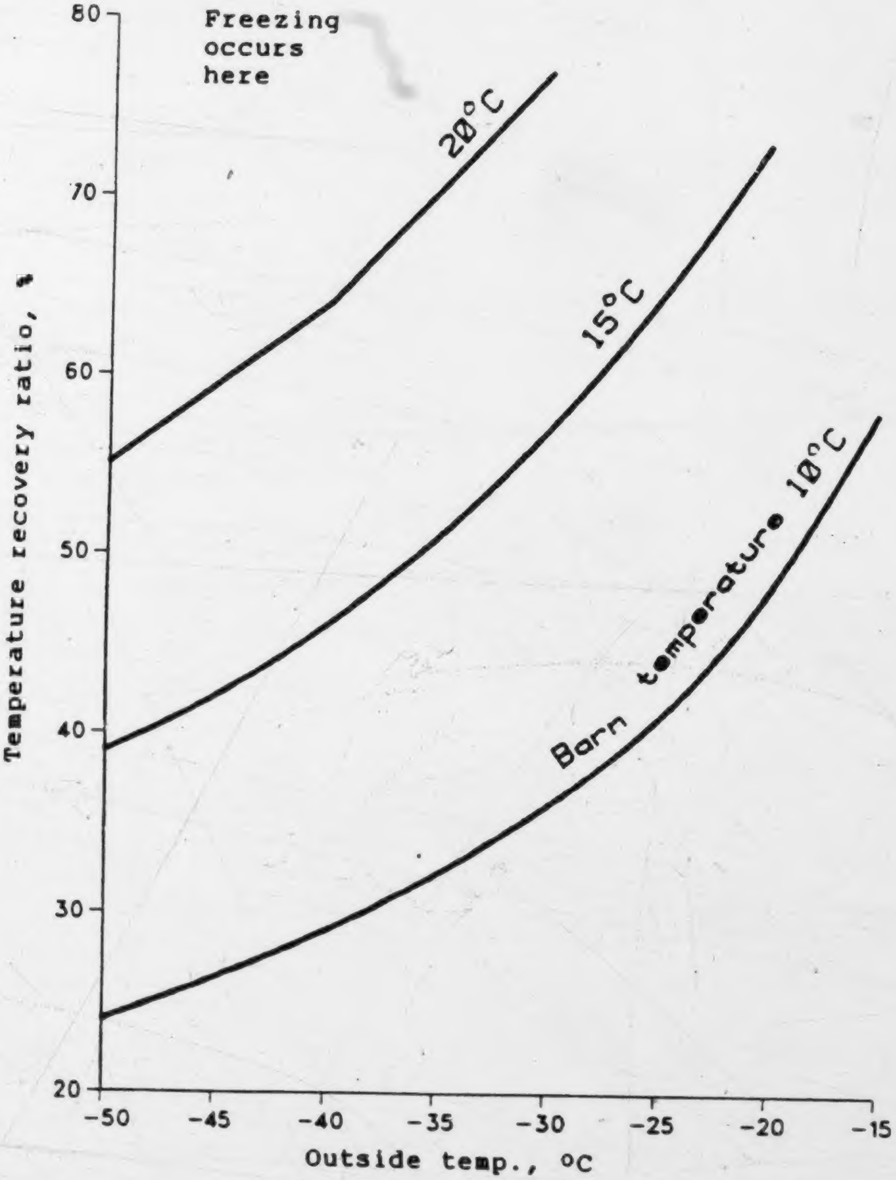


Figure 3.5 Desired TRR for Rh=75%

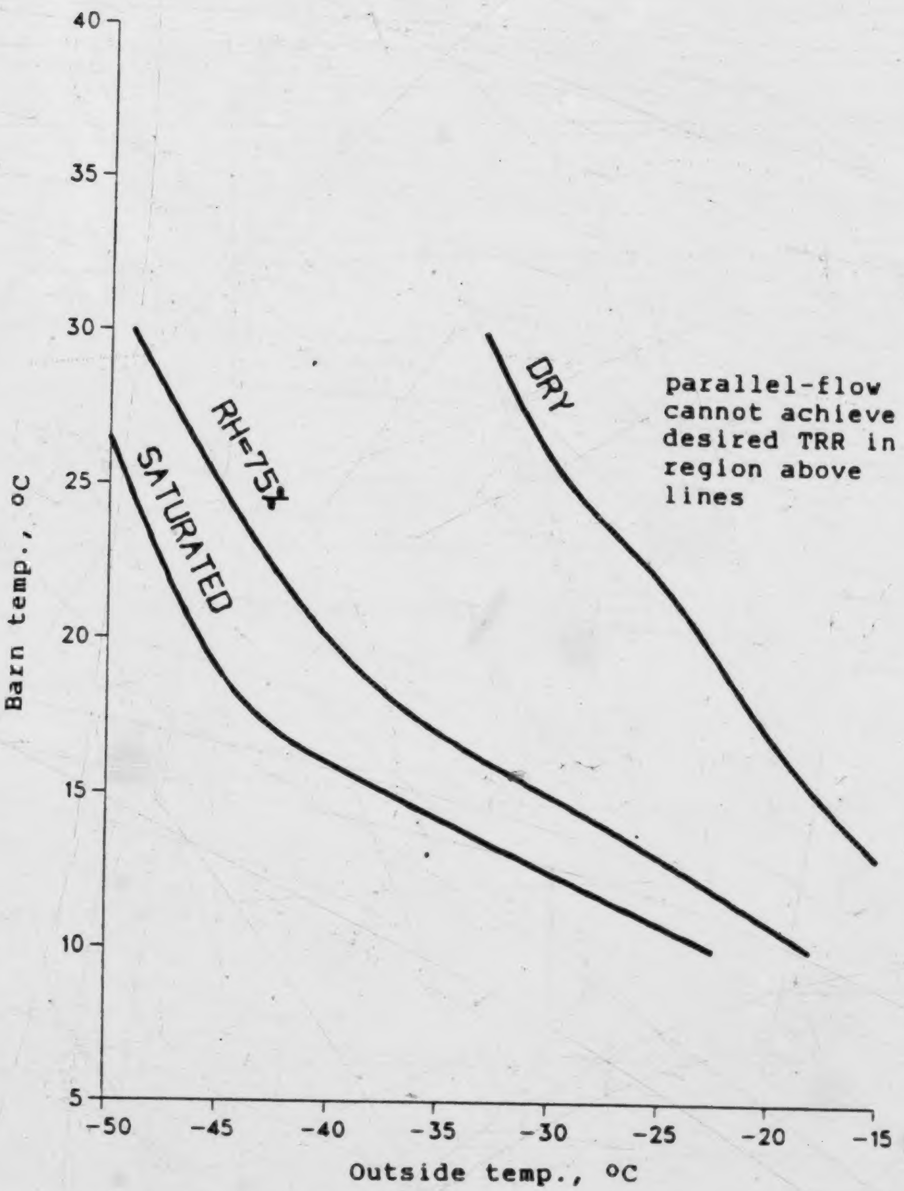


Figure 3.6 Limits for which parallel-flow heat heat exchangers can achieve the desired TRR.

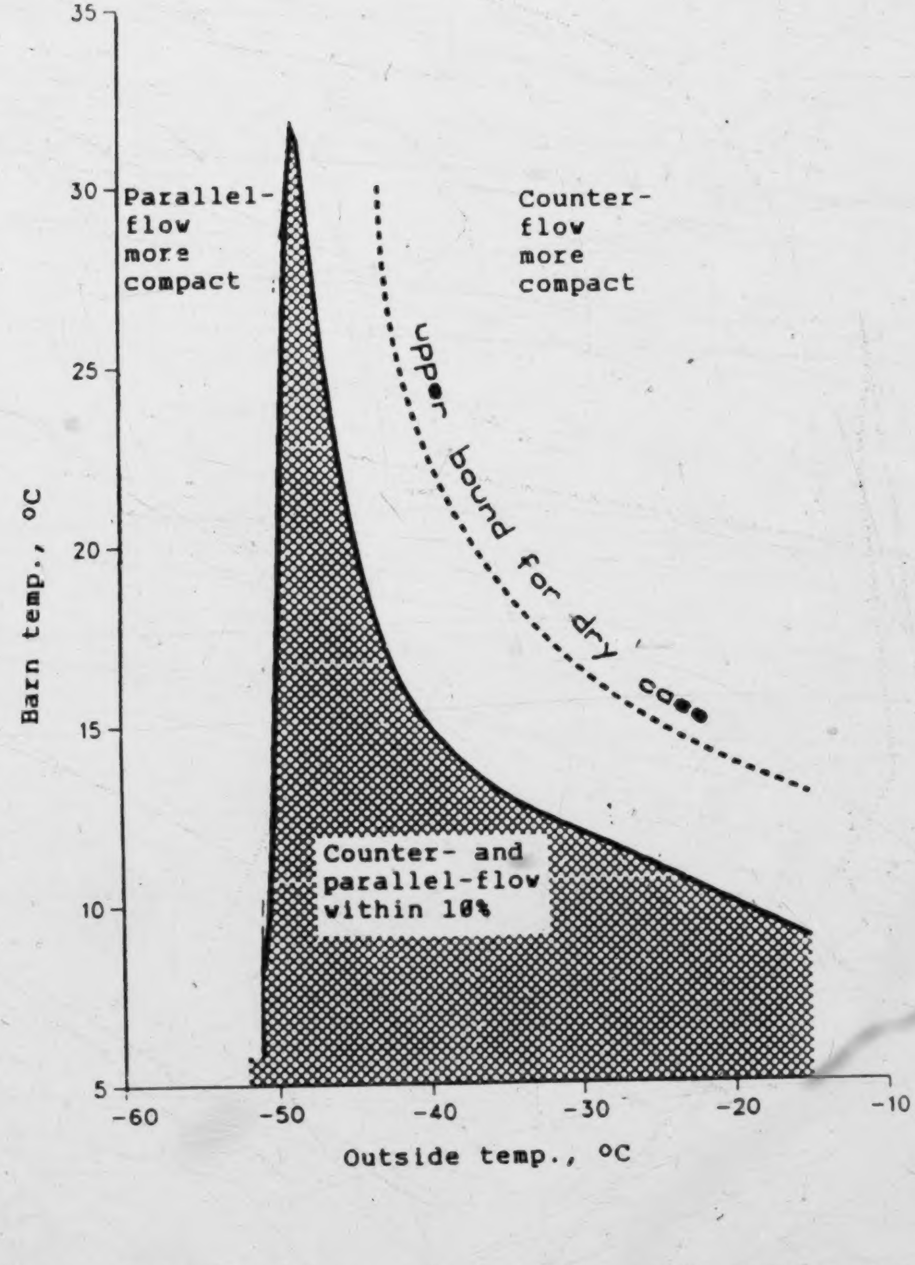


Figure 3.7 Comparison of relative size requirements for parallel- and counter-flow heat exchangers for achieving desired TRR, Rh=75%.

transfer area, while the region on the left is where parallel-flow are smaller. The dashed line shows where the upper bound of the shaded region would be for the case of dry exhaust air. Since parallel- and counter-flow are the extremes in temperature distribution, the area required for cross-flow designs to achieve equal temperature recovery ratios would also be within 10% of the other two for the shaded region, in addition to overlapping on both sides. Due to the non-uniform temperature profile in cross-flow patterns, icing can occur at temperature recovery ratios lower than that for either parallel- or counter-flow.

The assumption of a constant heat transfer coefficient along the length of the heat exchanger could produce a certain amount of error in the results as presented. Since heat transfer with condensation is higher than that for the unsaturated case, the heat transfer area required to lower the exhaust to its dew point should be greater than that determined by the computer models, whereas once condensation has begun, less heat transfer area should be required than that calculated, assuming the overall heat transfer coefficients are equal in both cases. For counter-flow, the lower heat transfer rate occurs where the temperature difference is also lowest; for parallel-flow the reverse is true. With that consideration, including variations in the value of the heat transfer coefficient along the length of the heat exchanger, the shaded area in Figure 3.7 would shift upwards and to the right (similar to the case of dry exhaust).

In summary, the conclusions reached by the modelling of the heat transfer process in parallel- and counter-flow directions were as follows:

i) Predicting heat transfer area using the LMTD equation can produce errors in excess of 10% due to the assumption of constant specific heat capacity used in the derivation of the equation.

ii) Counter-flow heat exchangers require less surface area than parallel-flow heat exchangers for most agricultural applications. For the cases of dry barn air and low barn temperatures, the area required for parallel- and counter-flow can be within 10% for equal performance.

3.2 Geometry Selection

From the review of the previous work on agricultural heat exchangers, the most promising choices for geometry were determined to be the flat-plate and the shell-and-tube designs. To aid in the selection of geometry type to be used in the prototype, another computer model was developed to calculate the annual operating costs for a particular design. By varying certain physical aspects of each design, the optimum combination was determined and then a comparison was made between these optimized designs.

The original choices for possible geometry type were as follows:

i) shell-and-tube: the exhaust air flowing through rigid PVC pipes with supply air flowing around them in a plywood shell.

ii) multiple-flat-plate: series of rigid polyethylene plates.

iii) polyethylene film: single polyethylene film seperating exhaust and supply air streams, encased in a plywood frame.

Each design was assumed to be adaptable to agricultural applications and, for modelling purposes, labor and maintenance costs were not included initially but were considered in evaluating the results. The choice for optimum design was based on capital cost of the materials and the fan power consumption.

3.2.1 Model Development

A BASIC program (Appendix B) was developed to calculate an annual operating cost for each geometry type based on an acceptable rate of return on material costs of 25% per annum and fan power costs assuming a fan efficiency of 33%, a running time of six months, and an electricity cost of $15/GJ. The cost of materials were based on University of Alberta Central Stores list prices, when applicable, or retail price from major hardware suppliers. The size of heat exchanger required was assumed to be one of equal supply and exhaust flow rates of $0.8 m^3/s$ and having a heat transfer area associated with 1.0 NTU. The heat transfer coefficient was assumed to vary with air velocity in a manner similar to the trend observed by Hodgkinson and Small (1984). By assuming the heat transfer coefficient was not dependent upon the geometry of the flow, a single equation could be used for all three designs. An equation was selected which was of a form resembling those for air in ASHRAE (1981) but which also roughly fit the graphical data presented in Hodgkinson and Small, resulting in equation 3.1:

$$U = 10\ v^{0.48} \qquad 3.1$$

where:

U = the heat transfer coefficient ($W/m^2 \cdot K$), and

v = the mean air velocity (m/s).

Pressure drops were assumed to be the result of entrance losses as the air enters a channel of the heat exchanger, skin friction losses according to numerical approximations for standard Moody curves based on the Reynolds number as the air travels through the channel, and sudden expansion losses as the air leaves the channel (ASHRAE 1981).

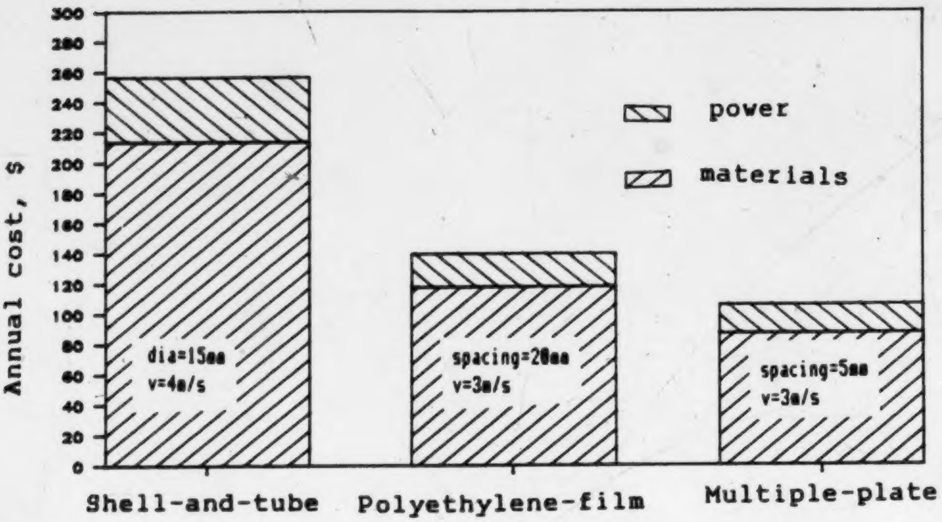
For the shell-and-tube model, the tubes were chosen to be standard PVC piping and their size was varied from 5 to 80mm diameter, and the air velocity was varied from 1 to 10 m/s. The arrangement of the tubes was chosen to be square with the distance between centers being 2.5 diameters. The shell side was assumed to be a plywood casing that enclosed the tubes on four sides. The fan power requirement for the shell side was assumed to equal that for the tube side, although it could be much higher depending upon the number of baffles used.

The multiple-flat-plate model was selected to represent a series of standard 1.6mm thick polyethylene plates at a spacing which was varied from 5 to 80mm and the air velocity was varied from 1 to 10 m/s. The entire series of plates were assumed to be arranged such that they would form a square cross-section. The plates were assumed to be enclosed in a plywood frame.

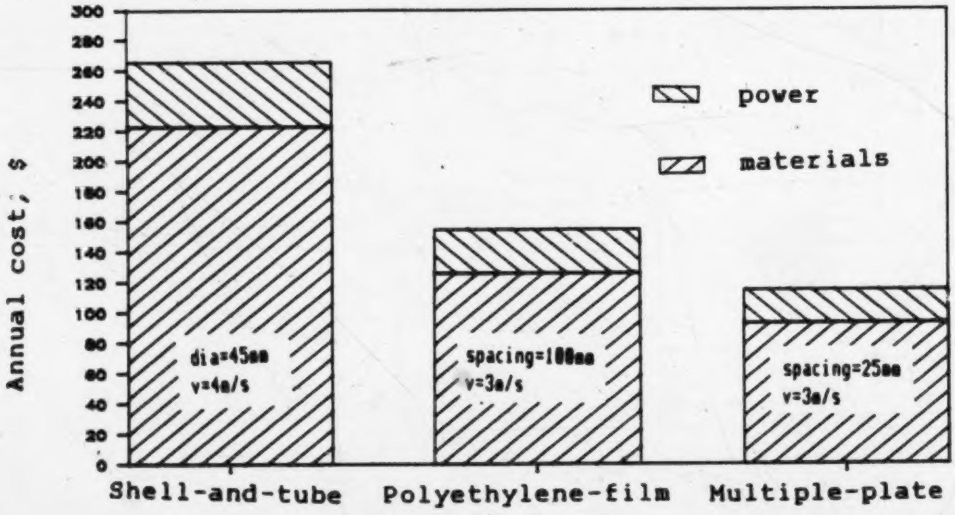
The polyethylene-film heat exchanger model was developed to be similar to the design of Hodgkinson and Small (1984). The simulated air flow channels were for a rectangular cross-sectioned plywood shell divided equally by a single sheet of polyethylene film. The width of the channels was varied from 5 to 160mm and the velocity was varied from 1 to 4 m/s (higher velocities were found to be undesirable due to fluttering of the plastic film).

3.2.2 Results

Figure 3.8 (a.) graphically presents the minimum annual costs for the shell-and-tube, multiple-plate, and single plate designs , as determined by the computer simulation models. In each case, the most economical design was one that was compact and had low air velocities. The resulting dimensions may not be practical for each of the three cases, however. For the shell-and-tube design, a tube diameter of 15mm may be usable, although dust problems could be



a) No size restraints



b) Modified for size restraints

Figure 3.8 Estimated annual costs for viable geometries.

encountered. In order to ensure problem-free operation, a diameter of 45mm would be more acceptable. The 5mm spacing calculated to be most economical for the multiple plate design would be similar to the design of Guertin et al. (1980). As noted in Chapter 2, a dust film of only 0.5mm on each plate would significantly restrict the flow (20% reduction in flow area for 5 mm spacing). A 25mm spacing would be more suitable for dusty environments. For the polyethylene film design, the 20 mm spacing calculated as being the most economical should not present problems with dust build-up. The flexibility and plasticity of the thin polyethylene sheeting would present problems in the maintenance of a constant 20mm spacing throughout the heat exchanger. The action of the air and the moisture in the chamber could cause the film to adhere to one side of the plywood casing. A spacing of 100mm would be more practical than the 20mm given by the program.

The computer models were modified to handle the above constraints on the dimensions and the program was rerun. The modified results appear in Figure 3.8 (b.). Assuming a labor cost of $10/hour and a 25% per annum allocation of the capital costs, the shell-and-tube design would have to require 40 hours less construction time to be as economical as the next lower cost design, with other costs being similar. Therefore, a reasonable conclusion would be that the shell-and-tube design is not as economical as the flat-plate designs as the total labor requirements for the

Hodgkinson and Small (1984) design were reported as 20 hours for construction and 2 hours maintenance per annum. The assumption of equal power requirements for both sides of the shell-and-tube design may have resulted in an underestimation of the annual cost since, for the shell-and-tube design tested by McGinnis et al. (1981), the shell side power requirement was 50% higher than that for the tube side (2.81 to 1.85 kW). If this high power is necessary to provide high velocities on the shell side for proper heat transfer, a more appropriate annual cost for the shell-and-tube design would be approximately 5% higher than that calculated.

The multiple flat-plate design appears more economical than the polyethylene film design in both the initial cost and power requirements. The total difference of $40 per annum could be outweighed by a longer construction time required for the multiple flat-plate design (16 hours) or a reasonable margin of error in the calculations. Although the shell-and-tube design is clearly less economical, the choice between the two flat-plate types should not be based solely on this simulation.

3.3 Energy Savings Through Heat Exchange

Heating requirements for most structures, such as houses, are calculated using the concept of the degree-day. When the ambient temperature is the same or above the minimum acceptable inside temperature of the structure, no heat is required. When the ambient temperature is below the minimum acceptable inside temperature, heat must be added at a rate approximately proportional to the temperature difference, according to the conduction, convection, and infiltration losses of the structure. This could be represented by the following equations:

$$q = 0 \; ; \text{ if } To > Ti$$

$$q = K\,(Ti - To) \; ; \text{ if } Ti < To \qquad 3.3$$

where:

q = heat required (kW),

K = constant of proportionality (kW/C),

To = ambient temperature (°C), and

Ti = minimum acceptable indoor temperature (°C).

If one chose a minimum desirable indoor temperature of 18°C, then the energy requirements, E, for a given time period would become:

$$E = JK\int(18 - To)\,dt \;\; ; \quad J = 0 \text{ if } To > 18°C \quad J = 1 \text{ if } To < 18°C \qquad 3.4$$

The average value of the integral in equation 3.4 for a given location over a one year period is a readily obtainable number referred to as the standard degree-days for that region. The heat energy requirement calculation then becomes:

$$E = K1 * DD \qquad 3.5$$

where:

K1 = constant K converted to appropriate units

(GJ/day°C)

DD = standard degree-days for the given location (day °C)

In the case of animal confinement housing units, the calculation of heating requirements cannot accurately be carried out using the standard 18°C degree-days since the heat produced by the animals can be sufficient to maintain the minimum acceptable inside temperature even though the ambient temperature may be considerably lower. A simple energy balance for a barn is represented by equation 3.6:

$$HP + SUPP = (Ti - To) * B + LAT + (Ti - To) * CV \qquad 3.6$$

where:

HP = total heat produced by the animals, litter, and other uncontrolled sources (W),

SUPP = supplemental heat required (W),

Ti = inside temperature (°C),

To = ambient temperature (°C),

B = constant representing total building shell losses (W/°C),

LAT = latent heat removed by ventilation, assumed constant (W), and

CV = ventilation sensible heat capacity rate (W/°C).

The amount of supplemental heat required will be zero until the ambient temperature drops to the point where the heat produced by the animals is no longer sufficient to maintain the minimum acceptable temperature. This limiting temperature is known as the heating deficit temperature, (T_{def}). The energy used for supplemental heating, E, can be determined using a modification of the standard degree-day concept. Equation 3.6 can be integrated, yielding:

$$\begin{aligned} E &= J \int[(T_i - T_{def} + T_{def} - T_o) * B + LAT + \\ &\qquad (T_i - T_{def} + T_{def} - T_o) * CV - HP]\, dt; \\ J &= 0 \text{ if } T_o > T_{def} \\ J &= 1 \text{ if } T_o < T_{def} \end{aligned} \tag{3.7}$$

Solving for the integral in equation 3.7 yields:

$$E = A(B+CV)(DD_{def} + Tx\ t_{def}) + A(LAT-HP)\ t_{def} \tag{3.8}$$

where:

E = annual energy requirement (J),

DD_{def} = degree-days for the deficit temperature (day°C),

$Tx = T_i - T_{def}$ (°C),

t_{def} = time that To is below T_{def} (day), and

A = conversion factor, (86400 s/day).

Similarly, the amount of heat recovered by a heat exchanger can be calculated by the following:

$$R = A*M_{max}*Cp*TRR*(DD_{def} + Tx*t_{def}) \tag{3.9}$$

where:

R = recovered heat (J),

M_{max} = maximum mass flow rate of the heat exchanger (kg/s),

Cp = specific heat of the air (J/kg°C), and

TRR = temperature recovery ratio.

Equations 3.8 and 3.9 require knowledge of the degree-day values for base temperatures below the standard of 18°C and also the time below this base temperature. Since these values are not readily obtainable, a computer program was developed to determine those for Alberta regions.

3.3.1 Model Development

A FORTRAN program was written to simulate the annual temperature variations for a given location (Appendix C). In simulating the temperature variations, the following assumptions were made:

i) maximum temperature (high) occurred at noon and minimum temperature (low) occurred 12 hours later for any given day.

ii) temperature varied linearly with time between the daily extremes.

iii) the temperature variation for each day was determined by the average high and low for that location.

iv) errors produced from using the simple linear model would average out over the course of a year such that the net error would be minimal.

As a data base, the average daily highs and lows for 22 Alberta locations, as supplied from tabulated values supplied by Environment Canada, were loaded into a computer file. The input parameters for the program were the desired location and the base temperature to be used. The program then read each day's high and low temperature for the entire year and, based on the above assumptions, numerically integrated the degree-days for the base temperature and summed the amount of time below that temperature.

3.3.2 Results and Discussion

The process of loading 730 data points for each of 22 locations could have been greatly simplified if the temperatures could have been obtained in a form directly readible by the computer, such as magnetic tape. In addition, the probability of human error is greatly reduced by a direct loading method, although no errors were detected in the course of verification of the results.

Table 3.1 shows a comparison between the results obtained from the program for one of the locations and published data (Thom 1954). A similar comparison for other locations and for the time below the base temperature could not be made due to the lack of information on those items. Figure 3.9 shows the result obtained for Edmonton and Table 3.2 and Table 3.3 presents the results for all 22 locations. The graph for degree-days is linear above base temperatures for which the maximum expected ambient temperature is always lower than the base. As the base temperature drops into the range of expected ambient temperatures, the curve becomes non-linear.

Although the differences between the published data and that produced by the computer program are all small (as

TABLE 3.1: COMPARISON OF DEGREE-DAY DATA PRODUCED BY MODEL AND PUBLISHED DATA FOR EDMONTON AREA.

Base temp. °C	Published data' (Thom, 1954)	Program	Difference (%)
18.3	5776	6095	5.5
15.6	4864	5192	6.7
12.8	4064	4343	6.8
10.0	3365	3610	7.2
7.2	2778	2968	6.8
4.4	2278	2387	4.8
1.7	1846	1908	3.3

' original data in °F

TABLE 3.2: DEGREE-DAYS FOR NON-STANDARD BASE TEMPERATURES, (DAY°C).

Location	Base temperature, °C					
	-15	-10	0	10	20	30
Athabasca	142	464	1710	3700	6760	10400
Banff	10	101	1100	3220	6470	10100
Brooks	49	203	1240	3090	5930	9490
Calgary	12	106	1030	2950	6010	9640
Claresholm	26	96	847	2650	5580	9180
Clearwater	58	208	1200	3340	6620	10300
Cold Lake	185	562	1850	3830	6860	10500
Coronation	84	373	1620	3590	6590	10200
Edmonton	104	387	1620	3610	6670	10300
Foremost	21	101	933	2730	5540	9090
Ft. Chipewyan	716	1280	2800	4990	8100	11700
Ft. Saskatchewan	94	358	1550	3470	6440	10100
Grande Prairie	134	452	1740	3740	6820	10400
Hinton	106	308	1340	3370	6540	10200
Kinsella	115	406	1630	3570	6570	10200
Lacombe	58	286	1430	3410	6480	10100
Lethbridge	14	50	785	2540	5390	8960
Medicine Hat	15	122	1020	2740	5440	8950
Peace River	239	643	2000	4040	7150	10800
Red Deer	62	308	1490	3470	6560	10200
Valleyview	107	365	1490	3420	6470	10100
Vegreville	168	523	1830	3840	6880	10500

TABLE 3.3: TIME BELOW BASE TEMPERRATURES, DAYS.

Location	Base temperature, °C					
	-15	-10	0	10	20	30
Athabasca	49	85	163	264	328	365
Banff	19	55	161	275	337	365
Brooks	40	60	150	242	307	365
Calgary	22	60	149	264	323	365
Claresholm	17	47	126	249	316	365
Clearwater	40	68	161	272	342	365
Cold Lake	57	90	166	255	326	365
Coronation	47	73	166	257	319	365
Edmonton	51	74	166	270	322	365
Foremost	14	53	138	240	308	365
Ft. Chipewyan	98	122	186	276	332	365
Ft. Saskatchewan	44	72	160	252	320	365
Grande Prairie	51	78	169	271	331	365
Hinton	36	68	156	271	326	365
Kinsella	44	80	160	258	320	365
Lacombe	47	64	157	267	320	365
Lethbridge	8	47	125	240	311	365
Medicine Hat	23	51	134	223	303	365
Peace River	62	97	175	277	326	365
Red Deer	47	66	164	270	328	365
Valleyview	39	70	149	270	326	365
Vegreville	55	84	170	271	321	365

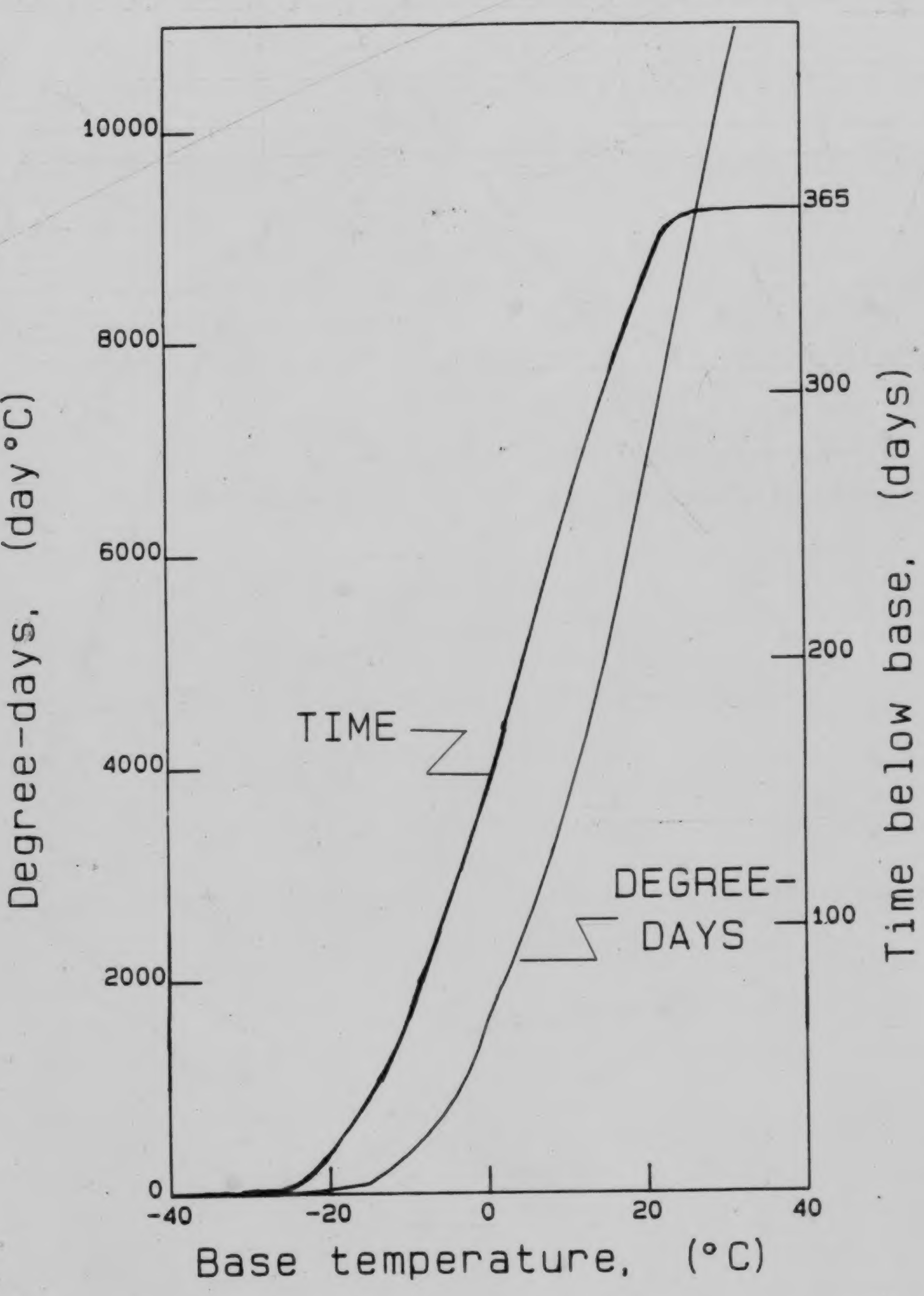


Figure 3.9 Weather data for Edmonton area

shown in Table 3.1) and acceptable considering the approximations involved in using the degree-day concept, in all cases the values produced by the program were the larger. This could possibly be an indication of the error produced by assuming the linear temperature variation with time between the daily highs and lows. Also of interest is that the slight difference between values becomes less at the lowest temperature range. For Edmonton, the value for degree-days calculated by the program becomes zero for base temperatures less than -25°C whereas the true number of expected degree-days should not reach zero until about 10°C lower (although no relevant published data were found). The error induced by representing the fluctuations in temperature by the average daily highs and lows therefore becomes more significant as the base temperature is lowered. An improvement in the model could be made by incorporating the standard deviations of the daily extremes as well as their means.

In conclusion, the computer model appears to yield good results for degree-day values for base temperatures near the freezing point and above, although it may slightly overestimate in this range. The results are not valid for base temperatures of -20°C or lower. Any heat exchanger installation being economically viable in a situation where the deficit temperature was -20°C or lower is not likely, so the limitations of the model should not restrict its usefulness.

4. DESIGN OF PROTOTYPE

4.1 Geometry Selection

The three designs used for comparison of operating costs in section 3.2. were the shell-and-tube, multiple-plate, and polyethylene-film designs, with the optimum annual operating costs calculated to be $266, $115, and $155 dollars respectively. On the basis of economics, the shell-and-tube design was not considered further as a possible choice for the prototype.

Although the calculated annual operating cost for the multiple-plate design was lower than that for the polyethylene-film design, the difference was not considered significant enough to eliminate the latter for purely economic reasons. For the purposes of this study, a design was desirable which could be fabricated and tested in a laboratory and then transferred to an appropriate animal housing unit for an in-field test. The optimum multiple-plate design, as determined by the program outlined in section 3.2, was a device approximately 0.7 x 0.7 x 2.5m, as compared to 0.2 x 2.0 x 20m for the polyethylene-film design. Due to the size of the latter design, disassembly would be necessary before changing location. The multiple-plate design would have a certain degree of portability. The smaller design would produce the lesser amount of inconvenience for the livestock operator and a greater degree of flexibility in installation method. Also, with respect to size, the computer model was used assuming flow rates of 800 L/s. If greater flows were required, the length of the polyethylene-film design required could necessitate major design modifications, depending upon the length of the barn used.

As determined from the literature review, by having the exhaust air flow vertically upward in the parallel plate design, the formation of the condensate would occur at the upper portion of the heat exchanger. The downward flow of the condensate produced by gravity would have the effect of washing dust off the heat transfer surface and lessen the frequency of required cleanings. This benefit could not be achieved with the polyethylene-film design because of the length of the exhaust channel, whereas it could easily be achieved with the multiple-plate design. As noted in the literature review, the lack of durability is a disadvantage of designs using polyethylene-film and uncontrollable factors such as mice and the need for inspections and cleanings could all reduce the productive life of any such device. The greater durability of the thicker rigid sheets in the multiple-plate design should give a longer expected life.

With respect to all the factors outlined above, the multiple-plate design has definite advantages over the polyethylene-film design, in addition to the calculated lower operating cost of the former. For these reasons, the final selection for type of geometry for the prototype was the multiple-flat-plate design.

4.2 Optimum Temperature Recovery Ratio

The temperature recovery ratio (TRR) of a heat exchanger is determined by the amount of heat transfer surface area used (assuming constant flow rates and heat transfer coefficient). To calculate the amount of heat transfer area to use, the following factors should be considered:

i) The amount of heat transferred per unit area of transfer surface decreases as the TRR is increased. The maximum TRR for a counter-flow heat exchanger is 1.0, which requires an infinite transfer surface area.

ii) From equation 3.16, the amount of energy recovered is directly proportional to the TRR of the heat exchanger.

iii) Cooling the exhaust air below the freezing point is undesirable.

iv) The maximum amount of heat recovery required is approximately the difference between the ventilation losses and the net heat production of the animals. Each of the above factors were considered to determine the desired TRR for the prototype and the methods used are outlined in the following sections.

4.2.1 Economic Analysis

Factors i) and ii) outlined in section 4.2. were combined to produce a simple model to estimate the optimum TRR based on the return on investment. The assumption was made that the TRR can be estimated from area using standard equations (ASHRAE 1981) and that icing problems can be ignored for the economic return. The cost of a 1.0m^3/s heat exchanger was estimated to be \$10 per m^2 of heat transfer area plus \$700 for fans and other installation costs. Power costs were determined as in the computer model in section 3.2.

Table 4.1 shows the expected rates of return for each \$100 invested into a heat exchanger for barns with recoverable heating costs of \$500, \$1000, and \$2000 per year. The rate of return was calculated assuming an expected life of five years with no maintenance costs, equal supply and exhaust flow rates of 1 m^3/s, and a material cost of \$10/m^2. The payback period was calculated assuming a straight-line payback of the investment by the net operating energy savings. For all three cases, the payback period reached a minimum level at a TRR of 0.5 and remained approximately equal at higher values of TRR. By noting the return on the last \$100 invested, however, for the case of low expected energy savings (as represented by the \$500 per annum case), it is not economical to build a heat exchanger above 0.5 TRR. As the recoverable energy costs increase, the maximum acceptable TRR increases, although the payback

TABLE 4.1: EXPECTED EFFECT OF HEAT TRANSFER AREA ON RETURN ON INVESTMENT.

			Recoverable heat/year ($)					
			500		1000		2000	
Invest. ($)	Elect. ($)/year	TRR	A (yrs)	B (%)	A (yrs)	B (%)	A (yrs)	B (%)
100	18	0.150	14.0	n.a.	6.1	n.a.	2.8	n.a.
200	20	0.265	8.0	35	3.7	93	1.8	208
300	22	0.358	6.4	24	3.0	71	1.4	164
400	25	0.434	5.7	16	2.7	54	1.3	130
500	27	0.497	5.4	9.5	2.6	41	1.2	104
600	30	0.551	5.3	5.0	2.5	32	1.2	86
700	32	0.596	5.3	0.5	2.5	23	1.2	68
800	35	0.635	5.3	-	2.5	17	1.2	56
900	37	0.669	5.4	-	2.5	12	1.2	46
1000	40	0.700	5.5	-	2.6	9	1.2	40
1100	42	0.725	5.6	-	2.6	3	1.3	28
1200	45	0.749	5.8	-	2.7	2	1.3	26
1300	47	0.769	5.9	-	2.8	-	1.3	18
1400	50	0.788	6.1	-	2.8	-	1.4	16
1500	53	0.805	6.3	-	2.9	-	1.4	12

A = payback period for the entire investment
B = rate of return for last $100 invested
n.a. = not applicable
- = loss on investment

period is the same as the case for 0.5 TRR. A heat exchanger with a TRR of 0.3 would be marginal for the cases of lower energy requirements.

4.2.2 Icing Limitations

As noted earlier, under certain conditions, the exhaust air may be cooled below the freezing point, causing the condensate to form ice and block or damage the heat exchanger. The TRR at which icing will first occur is dependant upon barn temperature, barn humidity, exhaust to supply air flow rates, and outside temperature. In addition, icing may occur at higher TRRs for cross-flow heat exchangers as compared to parallel- or counter-flow, due to the non-uniform temperature profiles across the exchanger in the former. The TRRs at which freezing first occurs in the latter two were calculated as a part of the program from section 3.1. These values are plotted for equal exhaust and supply flow rates and barn relative humidity of 75%, as shown in Figure 3.5. A barn with a 40% relative humidity and a temperature of 20°C has an air enthalpy of approximately the same as one with a relative humidity of 75% and temperature of 15°C, so the maximum TRR for the former case would follow that repesented by the middle curve. As can be seen from Figure 3.5, a heat exchanger with a TRR of 0.50 would require some form of protection from icing in many circumstances.

The results as presented would indicate that the assumption that icing problems could be ignored in the economic analysis in section 4.2.1. was incorrect. The effect would be that the savings associated with the higher TRRs were overestimated to a certain degree and lower rates of return would be more appropriate. In addition, to have a heat exchanger with a TRR low enough to eliminate the possibilty of icing would not be economical as indicated by the results presented in section 4.2.1.

4.2.3 Required Heat Recovery

The required heat recovery to eliminate all supplemental heating is approximately equal to the difference between the ventilation losses and the net heat production of the animals and other sources. The amount of this difference is a function of the type and age of the animals, the barn humidity and temperature, the outside temperature, and the ventilation rate. Using published guidelines for the minimum recommended ventilation rates and design heat production values (Hellickson and Walker, 1983), approximations were determined for the TRR required to maintain barn temperature without supplemental heating for various situations. The assumptions made were similar to those outlined for equation 3.13 with the exception that the conductive losses were assumed to be insignificant (as noted in the introduction, the heat lost through ventilation is much greater than conductive losses for most applications).

The results are presented in Table 4.2.

As seen in Table 4.2, heat exchangers should be unnecessary and therefore uneconomical in dairy operations. A deficit occurred only under the extreme outside temperature of -40°C and the high barn temperature of 20°C. Supplemental heat would not be needed since a drop in barn temperature would not be detrimental. Even if the high barn temperature was maintained, the high initial cost of a heat exchanger could not be recovered by energy savings.

In the case of swine, the efficient use of heat exchange could almost eliminate the need for supplemental heat, as indicated by the model. For the extreme case of -40°C, the size of heater required in a farrowing barn could be reduced by 60% with the use of a heat exchanger with a TRR of 50%. For feeder operations, the reclaimed heat at a TRR of 50% could reduce the size of heater by 75%. The smaller heater requirement would provide additional economic benefits to the straight energy savings.

The energy savings with broiler chickens would only be approximately 50% at best with heat exchange, although this would amount to the greatest dollar savings since the heat exchanger would be operating longer than for the other animal cases. As with the swine, the possibility of reducing the required heater size for new operations would have economic benefits.

TABLE 4.2: TRR NECESSARY TO RECOVER HEAT LOST THROUGH VENTILATION.

Animal type	Barn temp. (°C)	Outside temp. (°C)	TRR (%)
Dairy	20	-40	10
		-30	*
Swine			
- Farrowing (0 weeks)	20	-40	81
		-30	76
		-20	69
		-10	50
- Farrowing (8 weeks)	20	-40	21
		-30	5
		-20	*
- Feeder (20 kg)	20	-40	55
		-30	45
		-20	30
		-10	*
- Feeder (60 kg)	20	-40	67
		-30	59
		-20	47
		-10	21
Poultry			
- Broiler chickens (0.7 kg)	25	-40	100'
		-30	100'
		-20	100'
		-10	100'
- Broiler chickens (1.6 kg)	19	-40	80
		-30	75
		-20	66
		-10	46

* = no additional heat required
' = supplemental heat required

4.2.4 Selected TRR

By investigating the problem with icing, the possible savings with heat exchangers, the cost associated with greater heat recovery, and the energy requirements of various livestock operations, the optimum temperature recovery ratio would appear to be in the range of 0.45 to 0.55. The higher TRRs would be most economical in broiler applications, but may not be necessary since the periods when high heat recovery is needed are also the periods when operation would be limited by icing.

4.3 Material Selection

The most important initial criterion for selecting the material for the heat transfer plates is the cost per unit area. Strength, conductivity, fabrication method, and corrosion resistance are of lesser importance. As outlined in Section 2.2.1., the overall heat transfer coefficients were comparable for heat exchangers having polyethylene-film, sheet metal, and standard polyethylene-tubing heat transfer surfaces, suggesting that, in all cases, the thermal resistance of the transfer medium was negligible in comparison to those for the solid-gas interfaces.

The strength of the plates is of little importance as the stresses should be low. The static pressure difference across a plate would be approximately 100 Pa or less, resulting in insignificant loading in comparison to internal stresses resulting from fabrication, weight, or thermal expansion. A very brittle material could present problems during fabrication or very cold weather as failure of a plate would make its replacement difficult once construction was near completion.

Corrosion and low durability greatly affect the expected useful life of a heat exchanger. If the cost of the materials for a design with a long expected life was much greater than a less durable design, the frequent replacement of the latter would likely be more economical, although it would be less convenient and require more inspection.

Table 4.3 outlines the choices considered as possible materials for the heat transfer plates. The thickness of each material presented is that which is readily available and most cost efficient. Increased machining and merchandising costs can increase the price of thinner forms by 100%, even though less material may be present. Although there is a great difference in the thermal conductances between the materials, the greatest effect upon overall heat transfer coefficient which could be expected for the application at hand is approximately 10%. The greatest effect on heat transfer would be from the styrene and the lowest would be from the aluminum. In terms of cost per unit area, the acrylic and styrene are both more expensive than all the metal options considered, although corrosion resistance would favor the plastics. The two most economical options presented are the polyethylene followed by the PVC

TABLE 4.3: MATERIALS CONSIDERED FOR THE HEAT TRANSFER PLATES.

Material	Thickness (mm)	Cost ($\$/m^2$)	Thermal conductance ($kW/°C \cdot m^2$)
Aluminum	1.3	13.66	294
Cold roll steel	1.3	10.76	60
Galvanized steel	1.3	17.47	60
Acrylic	3.0	21.53	0.5
Styrene	3.0	17.70	0.4
PVC	1.6	7.85	1.3
Polyethylene	1.6	5.90	2.0

TABLE 4.4: CHARACTERISTICS USED TO SELECT MATERIALS FOR HEAT EXCHANGER CORE.

Property	Polyethylene	PVC	Better option
Cost ($\$/m^2$)	5.90	7.85	polyethylene
Thermal conductance ($kW/°C \cdot m^2$)	2.0	1.3	polyethylene
Thermal expansion (10^{-5} °C)	18	7	PVC
Strength (MPa)	20	50	PVC
Fabrication method	heat weld	chemical bonding	PVC

(polyvinyl chloride).

Table 4.4 is a comparison of the physical properties of polyethylene and PVC. Although polyethylene has the important advantage of lower initial cost and acceptable thermal properties and strength, it has very low adherence to glues and chemical bonding is not possible, making heat welding necessary for joining pieces. Special equipment and skill is necessary for the welding process and custom rates are approximately $3 per linear metre of weld. PVC can be bonded using solvents readily available at hardware retailers and it also has good adherence to other materials using contact cements. For the reasons outlined above, PVC sheeting was selected as the material for the heat transfer plates.

4.4 Design of Heat Exchanger Core

The basic form of the heat exchanger core is represented in Figure 4.1. The dimensions of the plates were largely predetermined by the selection of the flow rates, air velocity, and desired temperature recovery ratio. The design air flow rates were determined largely arbitrarily. The closer the flow rates to the minimum ventilation rate for moisture control, the more favorable the economic analysis, since fan and installation costs are not greatly affected by the flow rate, but energy savings can be considered proportional to the flow rate. In selecting a flow rate for the prototype, the desirability of a large

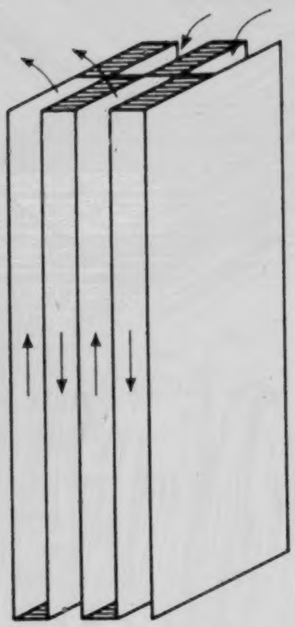


Figure 4.1 Basic form of a multiple-plate heat exchanger core.

unit was considered as well as the necessity of the unit to be adaptable to most existing operations, since the site of the in-field test had not yet been determined. The flow rate for the prototype, therefore, was selected at 800 L/s.

As determined by the computer program described in section 3.2 for the flat-plate design, the design which best satisfies both practical and economic considerations is one having an air speed of 3m/s and a plate spacing of 25 mm. Using a desired temperature recovery ratio of 50%, the resulting required length of heat transfer plate in the direction of flow is 2.1 m. If the external surface area of the design is kept to a minimum for insulative and cost reasons, the resulting dimensions of the device are 0.73 x 0.73 x 2.1m. Likewise, the dimensions for a 4 m/s design are 0.65 x 0.65 x 2.5m.

The PVC, from which the heat transfer plates were built, comes in the form of 1.2 x 2.4 m sheets. By splitting each sheet in half lengthwise, 27 of the resulting plates could be arranged such that the resulting dimensions would be 0.6 x 0.6 x 2.4m, which is close to the requirements for the 4 m/s design. To reduce the velocity to 3 m/s and maintain the desired TRR would require considerably more intricate construction or wastage of material, either of which would increase the cost more than that warranted by the fan power reduction. By revising the program in Section 3.2 to account for the increased cost produced by using PVC and an air velocity of 4 m/s, the predicted annual cost

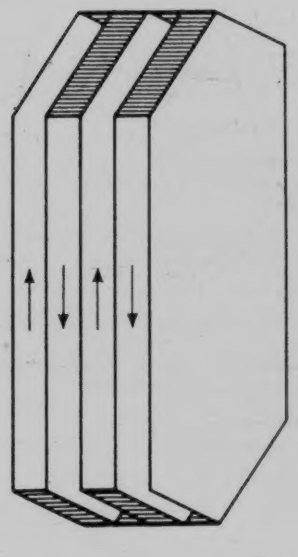


Figure 4.2 Modified plates

increased from $115 to $146 (which is still marginally below the $155 for the polyethylene film).

In order to facilitate the alternating of exhaust and supply air flows, to reduce cross-flow effects at the inlets, and to reduce stagnation regions, the rectangular shape of the heat transfer plates was altered to one having tapered ends, as shown in Figure 4.2. The alternate blocking of the air channels was planned to be achieved by using 72, 25mm wide strips of PVC sheet cut from the unused portion. As much of the sides as possible was planned to be left open to facilitate cleaning, observation, and instrumentation. Removable sides could then be attached to contain the flowing air.

4.5 Fan Selection

Heat exchanger performance and economic feasibility is greatly affected by the choice of fans used to move the air. The four parameters used to determine the effect of each fan on the heat exchanger economics were the initial cost, the flow rate, the fan efficiency, and fan power. Initial cost was determined by direct quotation from retail outlets. When available, the fan performance data were taken from independently published test results (PAMI), otherwise from supplied specifications. The source for any given fan is indicated in Table 4.5.

To determine flow rate for a given fan, the pressure losses through the heat exchanger were calculated from a

TABLE 4.5: FANS CONSIDERRED FOR PROTOTYPE.

Fan	Make	Model	Diameter (mm)	Source	Price ($)
A	EMI	WLA 4/50	490	PAMI 1983	424
B *	Seimens	400	400	man.	260
C	Seimens	450	450	man.	275
D	Majestic	16-A	400	man.	250
E	Pleasantaire	SW 14E1	355	man.	178
F	Pleasantaire	SW 12E1	311	man.	150
G	Penn-Breezeway	P-16Q	400	man.	325
H	Leco	A-1432	355	man.	341
I	Koenders	KS-18	450	PAMI 1983	145
J	Koenders	KV-15	368	PAMI 1983	165
K	Koenders	KV-18	450	PAMI 1983	175
L	Koenders	KV-20	500	PAMI 1983	200
M	Cyclone	60-AG	450	PAMI 1983	378
N	Ziehl-Abegg	451-6	450	PAMI 1983	306
O	Prairie-Pride	TR-12	311	PAMI 1983	395
P	Prairie-Pride	TR-18	464	PAMI 1983	450
Q	Prairie-Pride	TR-24	610	PAMI 1983	500
R	Blower	580	311	man.	250

man. - manufacturer's specifications
* - fan selected

TABLE 4.6: ESTIMATED FAN PERFORMANCE.

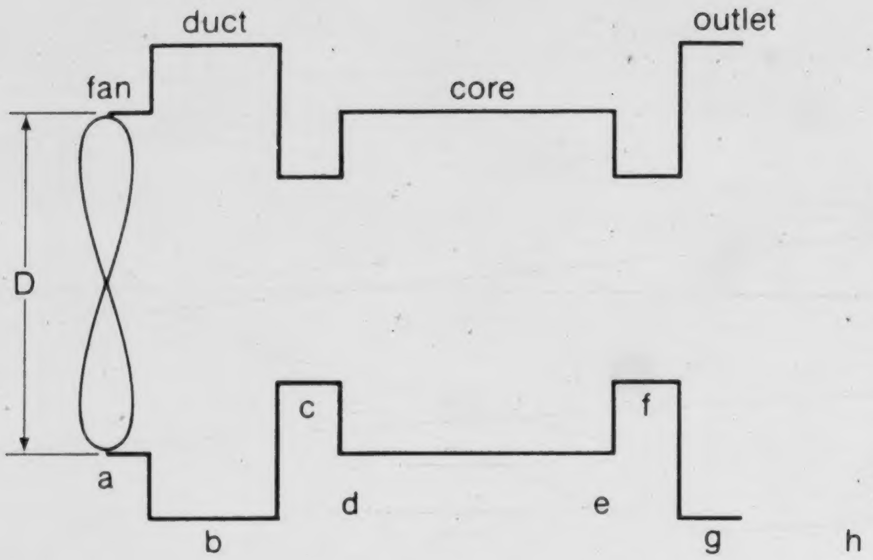
Fan	Flow rate (L/s)	Fan pressure (total,Pa)	Fan efficiency (%)	TRR
A	1500	100	42	0.36
B	* 875	50	44	0.42
C	1050	65	46	0.40
D	1100	80	27	0.39
E	950	80	20	0.41
F	720	60	25	0.44
G	1000	65	34	0.40
H	950	80	51	0.41
I	1120	75	22	0.39
J	700	40	8	0.44
K	1080	70	21	0.39
L	1580	125	33	0.35
M	1350	105	24	0.37
N	875	50	20	0.42
O	650	50	10	0.46
P	1280	90	25	0.37
Q	1300	85	27	0.37
R	880	95	10	0.42

* - fan selected

TABLE 4.7: ECONOMIC ANALYSIS OF FANS.

Fan		Electrical cost ($/yr.)	Rate of return (%) X	Y
A		154	68	3.1
B	*	43	60	4.7
C		64	68	6.3
D		140	69	2.5
E		173	66	(loss)
F		74	60	2.3
G		82	58	2.2
H		64	54	2.0
I		160	86	5.2
J		150	50	(loss)
K		175	78	3.7
L		250	95	4.7
M		250	60	(loss)
N		94	51	(loss)
O		140	27	(loss)
P		200	53	(loss)
Q		175	51	(loss)
R		360	37	(loss)

Energy lost through ventilation:
X - $3.00 per L/s
Y - $1.00 per L/s
* - fan selected



Cross sectional areas (m^2)

a - $\pi D^2/4$
b - 0.396
c - 0.150
d - 0.198
e - 0.198
f - 0.150
g - 0.396
h - ∞

Figure 4.3 Model for fan performance estimations

model as represented by Figure 4.3. The pressure losses through the system were considered to be comprised of the following steps (letters in parenthesis correspond to location in Figure 4.3):

i) (a-b) initial expansion from fan to duct,
ii) (b-c) contraction as heat exchanger is enterred,
iii) (c-d) slight expansion in heat exchanger,
iv) (d-e) skin friction along heat transfer plate,
v) (e-f) slight contraction at exit,
vi) (f-g) expansion to duct, and
vii) (g-h) final expansion out of system.

All calculations performed to determine the losses were as outlined in Daugherty and Franzini (1977) for incompressible flow in rectangular cross-sectioned channels.

All losses for the system are simply functions of the air flow rate, with the exception of the initial expansion loss, which depends on the flow rate and initial velocity. The total loss for the system can be reduced to a function of the flow rate and the fan diameter. The resulting equation for the geometry presented in Figure 4.3 is as follows:

$$Pt = 0.025\,(Q/D)^2 - 0.61\,Q^2/D + 64.2\,Q^2 - 9.38\,Q^3 \qquad 4.1$$

where:

Pt = total pressure (Pa),

Q = flow rate (m^3/s), and

D = fan diameter (m). Once the system response curve for a particular diameter of fan was determined from equation 4.1, it could be plotted against the performance curves of all fans of that diameter. The flow rate and associated pressure requirement for the fan were estimated by the point of intersection of its performance curve and the system response curve.

The TRR for a given flow rate can be estimated by using equation 3.1 to approximate the heat transfer coefficient of the heat exchanger. Then, since the mass flow rates are equal, that value may be used to calculate the TRR from standard equations for effectiveness (ASHRAE 1981), assuming the effect of condensation is within acceptable error. For the purpose of comparing fans, it was assumed that the cost of supplemental heat required would amount to $3.00 per L/s a year without any recovery. The rate of return on investment was then calculated according to the method used in Section 3.2.

Table 4.5 is a listing of the 18 fans considered for use in the heat exchanger. All fans were of the axial flow, propeller type with the exception of R which was a centifugal blower unit. For the purpose of using equation 4.1, the 18 fans were placed into one of six groups according to the fan diameter. These were the 610, 500, 450, 400, 355, and 311 mm sizes. Once the system curve for each group was determined, the expected performance for the fans was obtained, the results of which appear in Table 4.6. In determining the expected rate of return on the investment, the assumptions were made that the annual rate of depreciation was 20%, the electrical cost was $15/GJ, and the total cost of heat exchanger materials and labor was $800 (excluding fans).

Generally, the model predicted that the fans giving higher flow rates but lower TRRs would be more economical than the units moving less air but at a higher rate of heat recovery. Also, the savings on electrical energy from fans with high fan efficiencies were generally offset by increased initial costs for those units. A summary of the expected economic performance of the heat exchanger using each fan type is presented in Table 4.7. Included in this table is a comparison of the fans assuming an application where the gross energy savings by heat exchange would only be $1.00 per L/s. Under that condition, the units with higher fan efficiencies would become more favorable (units B and C for example). As noted in Section 4.4, if the flow rate of the heat exchanger was greater than the minimum winter ventilation rate, the energy savings are drastically lowered. For these reasons, the fan selected for the prototype was fan B, Siemens 400mm model, with a purchase price of $260 and an expected flow rate of 875 L/s.

4.6 Controls

As noted in earlier sections, during heat exchange the exhaust air may be cooled to the freezing point (or lower) when the outside air temperature is below a certain value. Dry air (RH less than 50%) is common to broiler operations (Feddes et al., 1984) and the associated lower apparent heat capacity of the exhaust air would make freezing in mid-winter more likely, although ice formation also is encountered with the moist (RH > 75%) environment of a swine unit (section 2.2). Ice removal is commonly performed by "defrosting" in which the cold supply air flow rate is reduced, stopped, or reversed. Reversal of the supply air flow would introduce the barn dust into the supply side of the heat transfer surface and thereby pollute an otherwise clean air passage. It should be noted that during the defrost cycle, the heat recovery is lowered to zero or, in the case of reversed supply flow, heat loss is increased above that for the minimum ventilation rates.

The mechanisms used for controlling ice build-up are as follows:

i) Manual inspection and removal. This method increases the work load for the operator and could result in long periods of down time or ineffective heat recovery. The coldest outside temperatures are typically at night, which is an inconvenient time for most operations. Some commercial units sold use this method.

ii) Timed defrost. The defrost cycle is activated regularly by a timer which is typically set to defrost 4 to 8 minutes every hour of operation. This amount of defrost would be greater than that required for ice prevention under most of the operating conditions, and may even be less than that required during times of very cold weather. A regular defrost cycle of 8 minutes per hour would reduce the effective TRR by that portion of time for which the defrost is activated (approximately 15%), thereby reducing the energy savings produced by the unit by that amount.

iii) Pressure controlled defrost. With this type of control, the defrost cycle is activated by the increased air pressure at the exhaust inlet resulting from the restriction caused by the ice build-up. The measurement can be made directly by a pressure transducer or indirectly by the increased power required by the exhaust fan. As noted earlier, the heat of fusion produced by the ice formation can be reclaimed if the ice leaves the system unthawed, although disposal of this ice could be a problem. If the exhaust air flowed upward, the ice would be prevented from leaving the system while still frozen, requiring total thawing and no net energy advantage of having the ice formed.

iv) Temperature controlled defrost. The temperature of the exhaust air as it leaves the heat exchanger is used to activate the defrost cycle through the means of a simple thermostat. The defrost cycle would only be used when required and, by setting the triggering temperature to just above freezing, ice should never be present in the system. Although seldom referred to in the reports of previous studies, this method appears to be an effective way of controlling ice formation (Larkin 1975,1979).

In view of the above points, the superior method of controlling the defrost cycle would be by temperature control. The desirable control device would be a thermostat with a remote sensor directly in the air flow path of the exhaust as it leaves the heat exchanger core. The control should be easily accessible to the operator to allow for slight ajustments to the set point. The reversal of the supply air would not be desirable, so the possible defrost cycles could consist of either stopping the supply air flow rate or lowering it to reduce the TRR to a level that would not initiate freezing. The motors of the fans selected for the prototype were adaptable to variable speed controls, so the latter option was investigated.

The standard control supplied by the manufacturer was a device that continuously regulated fan speed for an operating range of 10 to 30°C. That operating range was outside the useful range for defrosting. This mass-produced unit had a cost of $135. The benefit of a suitable variable speed control (provided one could be found) would be a slightly higher effective TRR than could be achieved by reducing the transient effects of the warming and cooling of the core produced by simply shutting off the supply flow. Assuming that a unit could reclaim $1000 per annum in energy, defrost would be required 5% of an average year, and variable speed could increase the effective defrost TRR by 5%, then the annual savings produced by the control device would be approximately $10. The actual value for the savings would probably be even less. The extra cost for variable speed controls would not be economical, especially if a custom made device would be required. Considering the above factors, the chosen method for activating the defrost cycle was a simple remote thermostat.

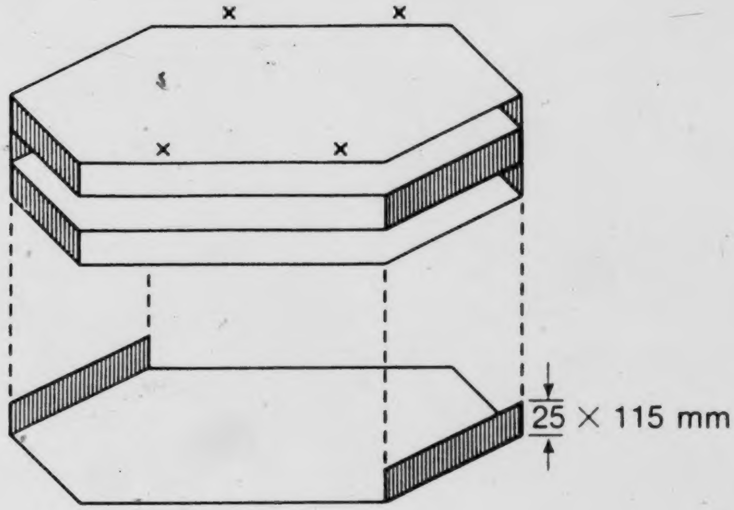
5. FABRICATION

In order to minimize initial costs, a heat exchanger design which could be fabricated with a minimum of required skills and specialized tools,would be desirable. Several techniques for constructing the prototype according to the proposed design were investigated.

5.1 Scale models

As an aid to conceptualizing the heat exchanger and possible ducting arrangements, a 1:10 scale model was built using 1 mm rigid cardboard (Plate 5.1). Through the use of model rectangular ducting, the heat exchanger appeared readily adaptable to various installations, as shown in Figure 5.1.

To investigate the possible problems in fabricating the prototype, another cardboard model was built with plates of 1:4 scale and full 25mm spacings between the plates. As found from this model, strips of PVC 25 x 460mm could possibly be glued between the plates to provide the alternate blocking of the flow channels needed to keep the two air streams separate, as in Figure 5.2. With the cardboard, the strength of the glued joint was sufficient to hold the plates apart at the ends without reinforcement. As became apparent, the lack of rigidity of the plates could produce problems in both the fabrication and handling of the prototype. The rigidity of the plates could be enhanced by inserting 25mm spacers down the entire 1685mm length of the



× - indicates location of 25 mm spacers required to keep plate apart

Figure 5.1 Blocking of the plate ends, cardboard model

Plate 5.1 1:10 scale model of heat exchanger core

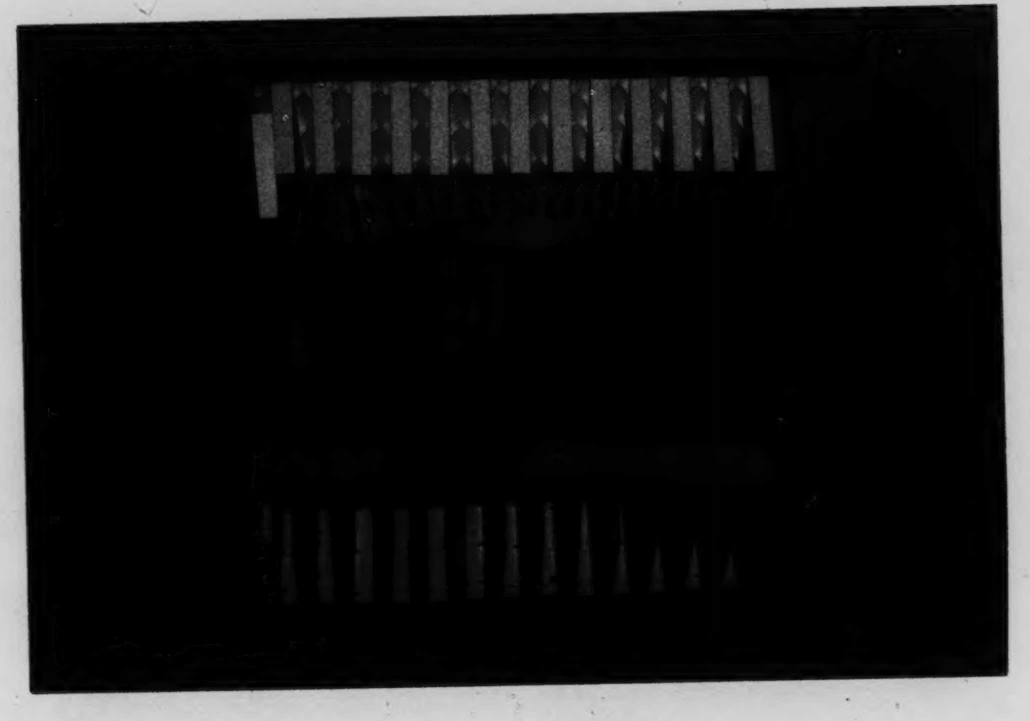


Plate 5.2 1:4 scale model of heat exchanger core

plates. The spacers could be either wood or PVC and would serve to maintain the correct distance between the plates. The result would be a rigid device with sealed channels for the air streams, thereby keeping leakage to a minimum. The disadvantages of such a system would be increased weight and required materials and, also, the sealing of the unit would restrict access to the area between the plates for visual inspection, cleaning, and installing of monitoring devices. An acceptable amount of structural rigidity in the model was achieved by using small cardboard spacers placed at intervals as shown in Figure 5.2 and Plate 5.2.

5.2 Prototype

The heat exchanger core was fabricated in the Agricultural Engineering workshop on the University of Alberta campus. In order to test the possible methods of holding the prototype together, a wooden framework was built to hold the plates in position as shown in Figure 5.3. The 27 plates were then cut from 14 sheets of PVC by first cutting the sheet in half lengthwise using a table saw, and then shaping the corners using a sheet-metal shear. Although not a necessary piece of equipment, the sheet-metal shear was used primarily for its speed. Care was taken to ensure the plates were as similar to each other as possible, thereby minimizing fabrication difficulties arising from irregularities in shape and size. The remaining one half sheet of PVC was cut into 52 strips 25 x 460 mm which were

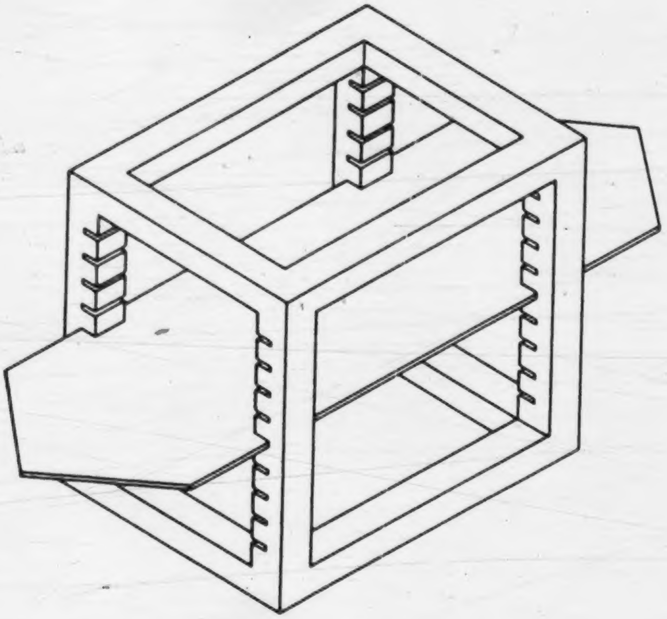
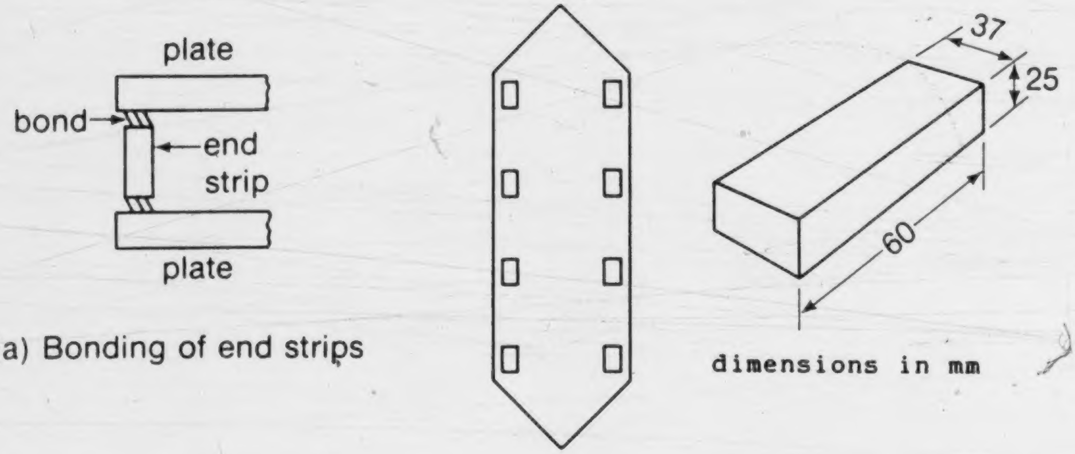


Figure 5.2 Test frame with one plate inserted.

used to alternately block the flow channels at the inlets.

The 27 plates then were guided into the wooden test frame and the end strips were glued into place. Two strips were glued at opposite corners of the first and second plate. Two more strips were glued at the other corners of the second and third plate and the process was continued upward for all 27 plates (as in Figure 5.2). A satisfactory bond was achieved using common PVC solvent (W1 by Schwartz Chemical), although the joint had to be held occasionally as the bond set (approximately 30 seconds). Best results, as determined by the time required for setting, were obtained by placing the joint between the edges of the strips and the faces of the plates (Figure 5.4 (a.)).

Once the end strips were in place, the plates did not provide sufficient rigidity to maintain the 25 mm spacing along the length of the core, as was predicted by the results of the cardboard model. As was noted earlier, the space between the plates should have been as accessible as possible, although a certain amount of accessibility had to be sacrificed for increased strength. The method used was to place small spacers at intervals as was shown in Figure 5.2 for the 1:4 scale model. The materials selected for the spacers were 25 x 37 x 60 mm wooden blocks, prepared by planing a standard 37 x 37 mm board to 25 x 37 and then cutting it into 60 mm lengths. The blocks were glued into position (Figure 5.4 (b.)) using a multi-purpose adhesive (DAP/Arno). After the manufacturer's recommended curing time



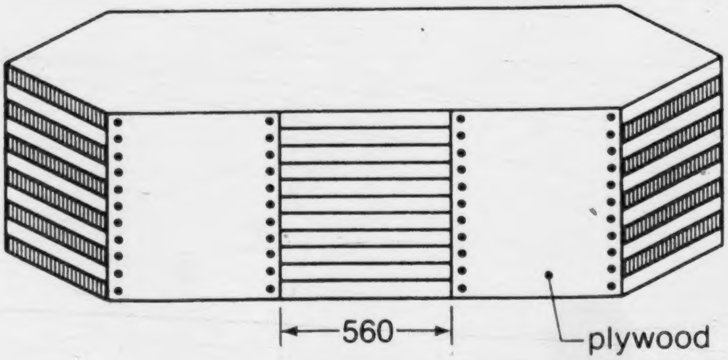
(a) Bonding of end strips

(b) Placement of wooden spacers for increased rigidity

Figure 5.3 Joining the plates

had elapsed, the wooden test frame was removed. The core originally maintained its shape, but after several hours deformations were observed as a result of failure in the wood PVC bond. The test frame was re-installed and the wooden spacers were re-set using solvent-based contact cement. The latter bond proved satisfactory.

At this stage the core had sufficient rigidity to rest unsupported on a horizontal surface. As the intended installation position was for vertically-upward flow of the exhaust air, additional strength was required. This strength was achieved by adding four 560 x 700mm pieces of 6mm plywood at the locations shown in Figure 5.5. At this time, 6mm foam carpet underlay was placed between the plywood and heat exchanger core to aid in the reduction of the possibility of leakage or short circuiting of the air streams. The plywood was attached to the wooden spacers using 30mm spiral nails. The addition of the plywood supports further decreased the accessibility of the core, although an opening of 560mm remained in the central portion of the heat exchanger (Figure 5.5). The core then had enough durability to allow for transport by truck for the necessary laboratory testing.



dimensions in mm

1

Figure 5.4 Plywood supports

6. EXPERIMENTAL FACILITIES AND PROCEDURES

6.1 Laboratory testing

Testing the heat transfer properties under laboratory conditions was not considered. As noted in earlier chapters, the effects of dust and humidity in livestock environments make standard laboratory tests of heat exchangers unreliable in predicting field results. Data from fan tests would also not be directly applicable to field tests due to flow rates in the field being affected by differences in ducting, barn pressure, thermal effects, and dust build-up. However, some useful insights were gained by the laboratory fan tests such as locating possible leakage sites and determining structural stability. A series of runs were made in which air was drawn or blown through the heat exchanger core. By varying the degree to which the sides were sealed, an estimation of expected leakage rates could be determined for that particular set-up.

6.1.1 Location

The initial air flow and fan performance tests were performed in the Agricultural Engineering Harvesting Laboratory at the University of Alberta Ellerslie Research Station. The fabrication of the fan and core test housing, the post-laboratory test modifications, and the fabrication of the final heat exchanger housing were also performed at that site.

6.1.2 Test equipment and procedure

A structure for housing the fans during testing was constructed in the form of a 720 x 720 x 2400 mm rectangular duct using 12 mm particle board. As indicated in Figure 6.1, the fans were installed so that one was blowing the air through one channel using positive pressure and the other was drawing air through the opposite channel using suction. A series of three runs measuring air flow rates were performed. The first run was performed with the sides of the heat exchanger core left open between the plywood supports. The second run was performed with the side openings sealed with plywood sheets. The third run was performed with the side openings sealed with 16 mm foam carpet underlay acting as gaskets between the plywood and the sides, as was used when installing the plywood supports. The air flow rates were measured at the end of the heat exchanger opposite the fans. The flow rate was determined by measuring air velocity at 6 locations for each of the 13 openings for a particular side. The measurements were made with a hot wire anemometer (Sierra Instruments).

The fans were then attached to a rectangular discharge duct constructed for previous tests (Feddes et al., 1984). The flow straightener within the duct produced a near-uniform velocity profile at the outlet, providing an indication of the fan output with no leakage.

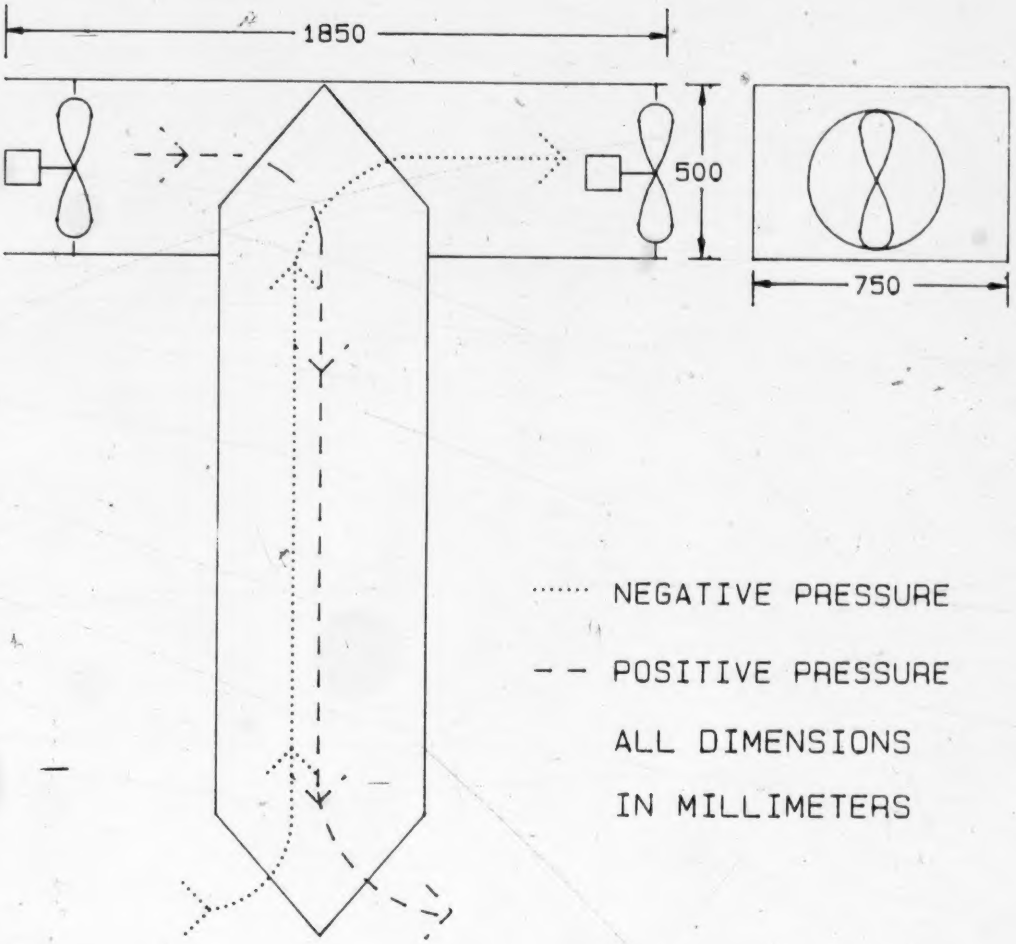


Figure 6.1 Ducting for fan tests

6.1.3 Results

The summarized results of the lab fan tests appear in Table 6.1. The large difference in flow rates for each run could be assumed to be the result of leakage, presumably through the sides of the heat exchanger core. When attached to the discharge duct, the output of the fans were measured and found to be 810 L/s, with approximately the same static pressure produced (30 Pa) as the fan blowing the air through for run 3. This would tend to indicate that the amount of leakage for the positive pressure side was within acceptable limits (less than 5%) when the foam gaskets were used. Without the use of the foam, the leakage was unacceptable (approximately 30%). Surprizingly, the leakage without sides was only 50% for the positive pressure side. Leakage was much greater for the negative pressure side, ranging from 75% for the case of no sides to 30% with the foam gaskets.

The results indicate that for the case of little or no pressure differential between the inlet and outlet of the heat exchanger, leakage and consequent mixing of the air streams can be minimized by blowing, as opposed to drawing, the air. Leakage was greatly reduced by using foam gaskets to aid in the sealing of the sides of the heat exchanger.

6.2 Field testing

TABLE 6.1: LABORATORY TEST FOR LEAKAGE OF CORE.

Run	Negative pressure Flow rate (L/s)	Positive pressure Flow rate (L/s)	comments
1	170	420	open sides
2	320	560	plywood sides
3	570	780	plywood & foam

6.2.1 Location

The fabrication of the heat exchanger housing was delayed until the exact location of the field test was determined. The site selected was a broiler barn near Morinville, Alberta used in previous studies (Feddes et al. 1984). The housing unit was a 12 x 46m wood-framed broiler barn. Ventilation was provided by seven two-speed exhaust fans situated along one side of the building, and a continuous-slot, fresh air inlet under the eave of the opposite side. The total barn ventilation rates, as measured in the tests by Feddes et al. (1984), ranged from 750 L/s to 4332 L/s depending upon the size of the birds housed. The expected ventilation rate of 800 L/s for the heat exchanger should not have exceeded the minimum at any time during the period of occupation.

Heating of all barns on the site originated from a central coal-fired boiler unit. Hot water was circulated through 51mm internal diameter, black-steel pipes mounted horizontally on the inlet wall of the barn.

The original duration of the test period was selected as the complete growth cycle for one batch of broilers (approximately 7000 birds). The test was later extended to

TABLE 6.2: FIELD TEST SCHEDULE.

Run	Date	Age of birds (days)	Approx. ambient temp (°C)	Number of data sets
1	25/11/85	16	-27	47
2	29/11/85	20	-20	49
3	6/12/85	27	-5	50
4	19/12/85	40	1.5	44
5	29/01/86	18	-10	46
6	2/02/86	22	4	42
7	19/02/86	39	-30	30
8	20/02/86	40	-20	149
9	24/02/86	44	5	5

cover three growth cycles, which translates into a test period of 24 weeks.

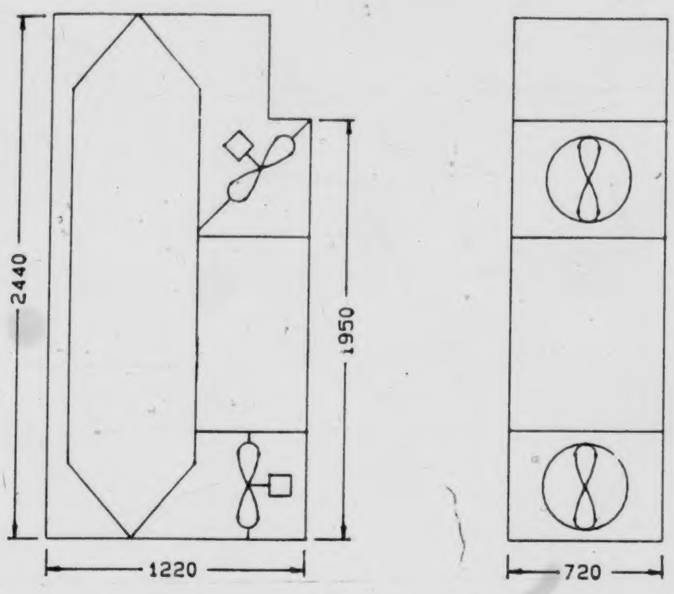
The possible methods of installing the heat exchanger are shown in Figure 5.1. Since the field test involved a temporary installation, the preferred method would be one that involved the least amount of physical alteration to the housing unit. In addition, since the heat exchanger was to be monitored, accessibility was another desired feature. Ceiling mounting would be the least favorable method considering the previous two factors. Minimal alteration of the building was achieved by installing the heat exchanger in an existing 2000 x 900 mm doorway, one third of the distance down the barn on the fresh air supply side.

The housing for the heat exchanger core was designed so that the entire unit could be installed into the doorway or removed quickly. If a problem arose when the barn was occupied with birds, quick removal and reinstallation of the original door would minimize the exposure of the birds to the outside. The housing as designed is shown in Figure 6.2

and as fully installed in Figure 6.3 and Plate 6.1. Parallel-flow was the easier flow direction to accomodate, therefore that was the pattern selected.

The fans were installed so that the exhaust air was forced out under positive pressure while the supply air was drawn through under suction. In the laboratory tests, excessive leakage was found when negative pressure was used, but it was used again in the field test to determine whether leakage could be reduced by the negative pressure gradient from the ambient to barn air. Negative barn pressure was expected since the required ventilation rate for the barn was usually much higher than the output of the heat exchanger. The open sides of the core were sealed with plywood with the foam gasket for the face inside the barn and 50mm styrofoam with the foam gasket for the side exposed to the outside. The entire housing was covered with the 50mm styrofoam to prevent excessive heat transfer directly out of the heat exchanger unit.

As exhaust air is cooled, the air becomes saturated and the moisture condenses out. Since the exhaust was to flow in an upwards direction, the condensate would flow back down to the bottom of the heat exchanger core. From the data supplied from Feddes et al. (1984), the highest expected moisture content for the barn would be \0.008 kg/(kg dry air), approximately half of which would condense out at 0°C. For an airflow rate of 800 L/s, 13 L of condensate would be formed per hour under the above conditions. In order to



ALL DIMENSIONS
IN MILLIMETERS

Figure 6.2 Heat exchanger housing

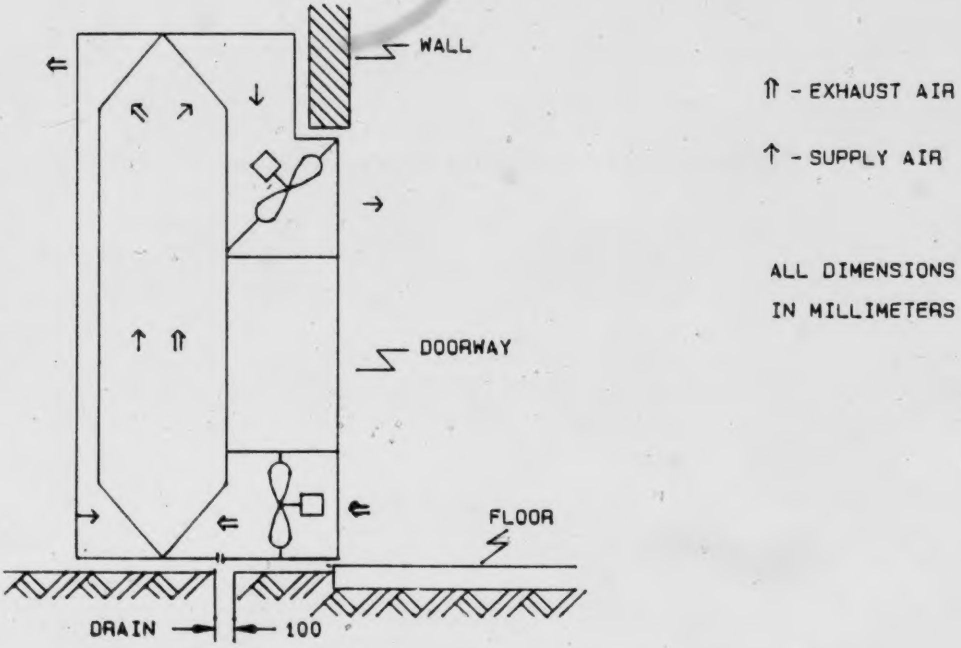


Figure 6.3 Heat exchanger installed

Plate 6.1 Heat exchanger installation

handle this water, a drain hole was drilled beneath the heat exchanger, as shown in Figure 6.3. A 50mm hole was cut into the bottom of the plywood heat exchanger housing through which water could pass into the drain. A 2m heat tape was placed into the drain hole to ensure the ground in the region of the drain did not freeze entirely and prevent the moisture from seeping away.

6.2.2 Controls

As outlined in Section 4.6, the selected mechanism for activating the defrost cycle was a thermostat monitoring the exhaust outlet temperature. The device selected was a standard, double-throw thermostat with remote sensor which had an operating temperature from -30 to 30°C. The sensor was attached at the center of one of the exhaust outlets. The control was fixed to the side of the heat exchanger housing where it could be adjusted easily by the operator. The differential (reset) value was adjustable between the range of 2 to 6°C and was initially set at 2°. No further adjustment was found to be necessary. The current to the fans could be interupted through the use of two single-throw switches. A sketch of the control layout appears in Figure 6.4.

6.2.3 Data collection equipment and procedure

The performance of the heat exchanger during the field test was measured and the data collected by the mobile

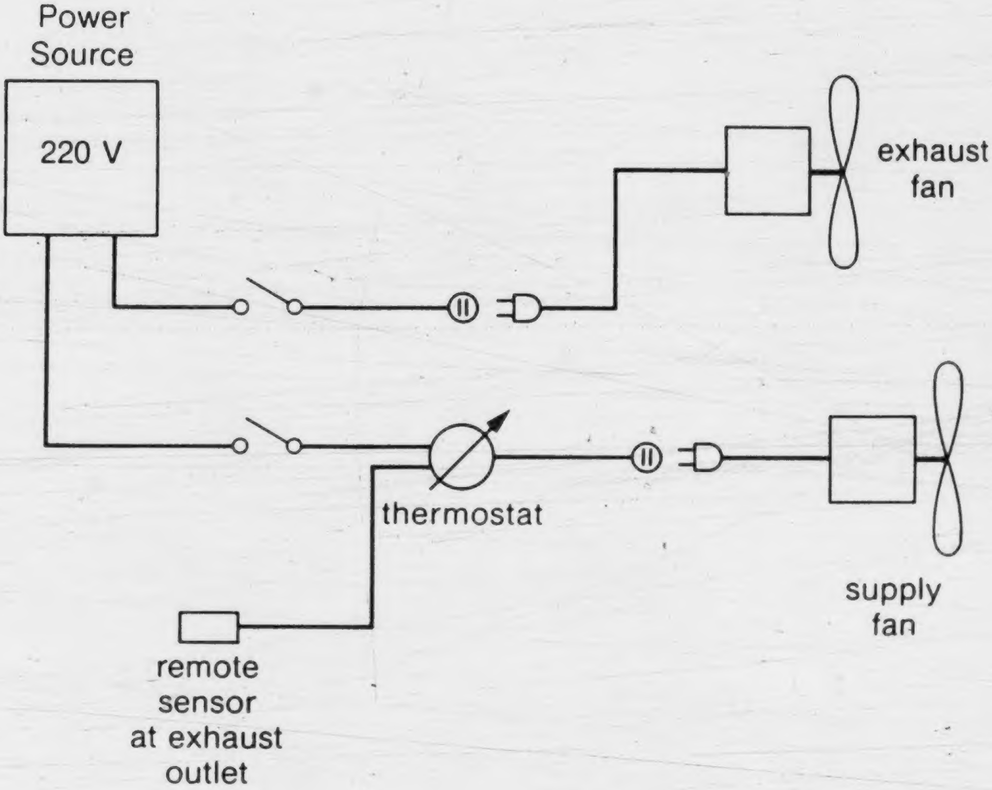


Figure 6.4 Schematic of fan wiring

environmental data aquisition system of the University of Alberta Department of Agricultural Engineering as described by Feddes and McQuitty (1981).

Air temperatures were measured by 26 thermistors, placed along the supply and exhaust flow paths. Air samples were drawn from six locations from which moisture levels were determined. A diagram of the thermistor and air sample tube locations is shown in Figure 6.5. Barn dust levels also were measured.

The data aquisition system monitored the thermistor readings every 20 minutes and determined the dew point at each air sample position twice per hour. The analog voltages were digitized by the system and stored on standard floppy diskettes. FORTRAN and BASIC computer programs were written to convert the digital information into more readily understood forms (Appendix D).

The heat exchanger was installed into the broiler barn on November 8, 1985, two days before the arrival of the first batch of birds. It was first turned on on November 25, 1985 when the birds were 16 days old. On November 26, the heat exchanger was turned off because the heating system of the barn was not able to maintain the required barn temperature. The contributing factors for the failure of the system were as follows:

i) small size of the birds

ii) smaller than normal batch size (7000 birds instead of the normal 8000)

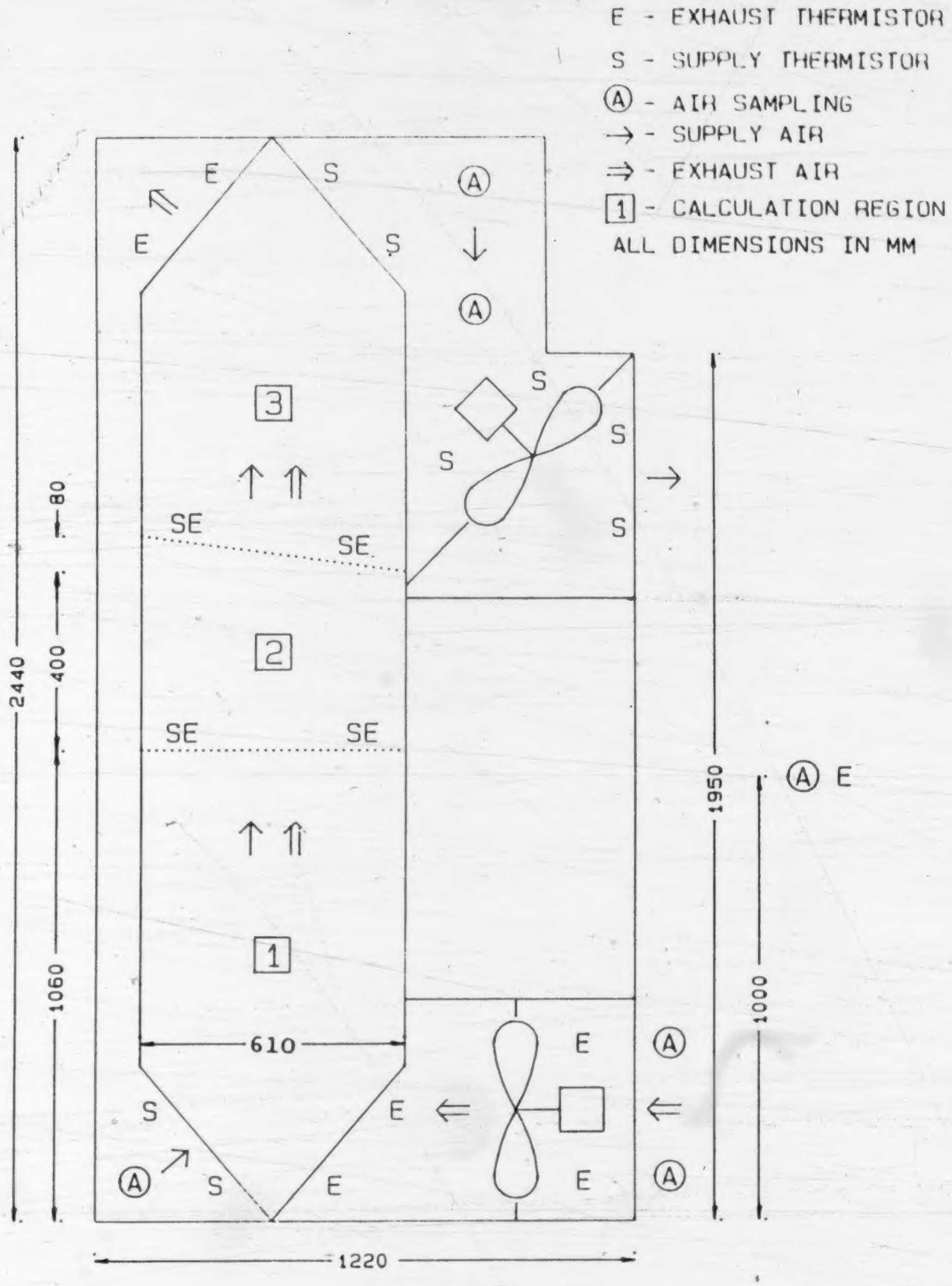


Figure 6.5 Thermistor and air sample locations

iii) low outside temperature (-27°C)
iv) strong winds and high infiltration rate
v) partial failure of the central coal-fueled furnace.

The problems with the heating system were corrected on November 27 and the heat exchanger was restarted on November 28. The heat exchanger operated continuously until December 23 at which time the growing cycle of the birds was complete. It remained installed in the doorway but all inlets and outlets were sealed with fibreglass insulation while the barn was empty to prevent any unnecessary ventilation losses from the building. The barn was filled with a second batch of birds on January 12, 1986 and the heat exchanger was started on January 28. It remained running continuously until the end of that growth cycle which was February 25, 1986. The total time the heat exchanger operated at that installation amounted to 53 days.

The schedule of data collection sessions appears in Table 6.2. The dates were selected so as to obtain runs for a variety of ambient temperatures and ages of birds. The data collection sessions averaged approximately 24 hours in length, with the exceptions of run 8, which was 60 hours, and run 9, which was performed for only 1.5 hours before the system was removed from the test site. Air flow measurements were made periodically and, on one occasion, the protective grille of the exhaust fan was cleaned of debris which could have interfered with performance. The dust inside the heat exchanger was inspected regularly, but care was taken to minimize the disturbance of the growth of the dust film build-up on the PVC plates. Power consumption of the fans was measured at the beginning of each run by placing an ammeter in the circuit.

6.2.4 Modifications

Over the duration of the field test, several modifications to the original design became necessary, in addition to minor maintenance requirements. The changes made were as follows:

i) December 6, 1985 - Styrofoam insulation was added to the barn side of the heat exchanger housing. A layer of 4mm polyethylene film was inserted between the styrofoam and the foam gasket at the inspection site. These changes were made in response to the discovery that leakage was occurring from the exhaust to the supply air streams.

ii) December 19, 1985 - The supply fan was moved to the supply inlet (Figure 6.6) putting both sides of the heat exchanger under positive pressure. This was done because the changes of December 6 were not totally effective.

iii) December 19, 1985 - The grill of the exhaust fan had become partially blocked by feathers, dust, and straw. The grill was cleaned and the effect noted on fan power consumption and air flow rate.

iv) February 19, 1986 - A rectangular duct was added at

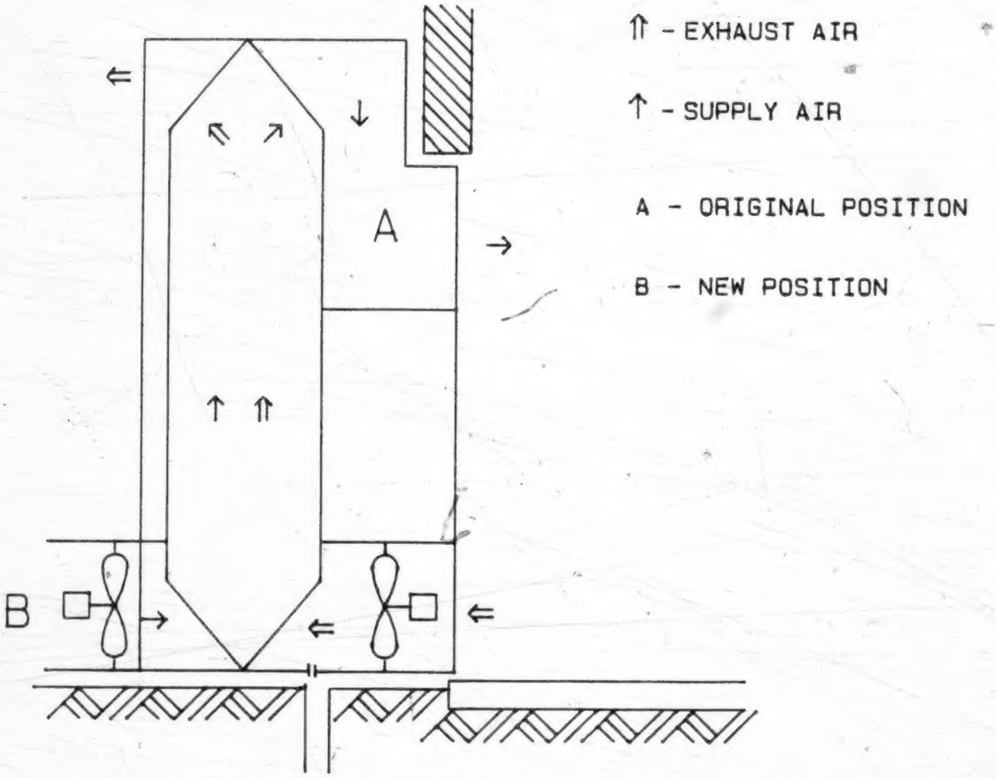


Figure 6.6 Supply fan relocation

the supply outlet to redirect the supply air upward. This was done to increase the mixing of the supply with the barn air as it entered the barn. Without the duct, the incoming air was flowing downward to the floor of the barn when ambient temperatures were very low.

v) February 24, 1986 - Strong winds had blown part of the styrofoam insulation off of the outdoor portion of the heat exchanger. This was repaired.

7. TEST RESULTS AND DISCUSSION

7.1 Heat exchanger performance

A listing of the programs used to calculate the operating parameters of the heat exchanger appears in Appendix E. The input files containing the data were the output files from the programs in Appendix D.

7.1.1 General performance

Condensate formed on the heat transfer plates and flowed to the bottom of the heat exchanger. The drain carried the water away satisfactorily, although on one occasion feathers and dust plugged the 50 mm hole in the bottom of the housing. The blockage was caused in part by the heat tape cord passing through the hole as well. The condensate pooled in the bottom of the housing and leaked into the barn through the fan casing. Once unplugged, the problem was not re-encountered.

The defrost control mechanism performed satisfactorily, as no icing occurred at the exhaust outlet. The thermostat was bypassed for a four hour period when the ambient temperature was -35°C and ice formed at the exhaust outlet to the point of nearly blocking the flow in some places. When the thermostat was reintroduced to the circuit, the ice was gone in 30 minutes. A graph of the outlet exhaust temperature for -35°C weather is shown in Figure 7.1, showing the typical defrost cycle at that temperature.

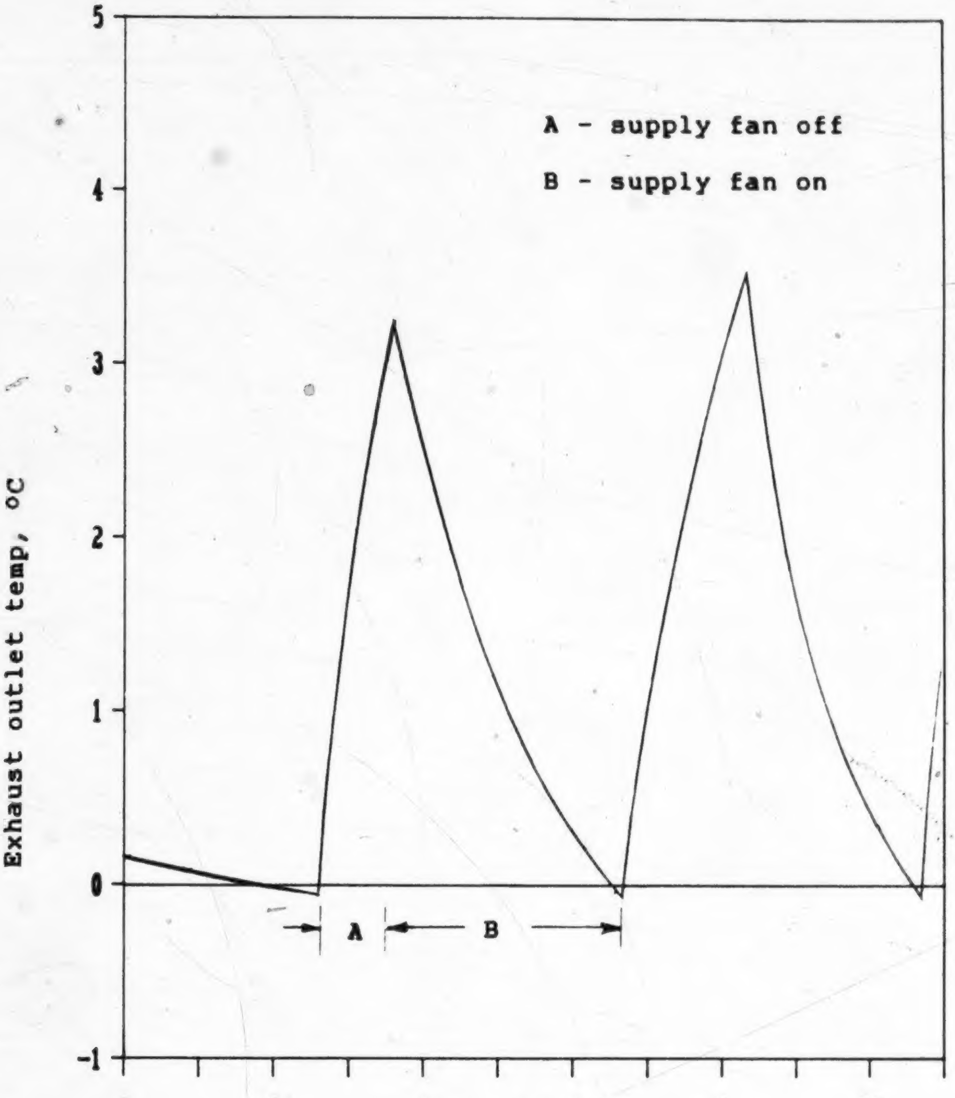


Figure 7.1 Exhaust temperature showing defrost cycling

Table 7.1 shows the measured dust concentrations in the barn for each of the runs. Problems with the data aquisiton system led no data for some runs. There was no noticeable variation in dust build-up along the length of the plates. Dust could be seen in the exhaust as it left the heat exchanger unit and deposits were visible on the snow near the outlet. Considerable amounts of dry dust left the heat exchanger core when the system was restarted for the second batch of birds, after being off for 36 days. The 25mm spacing appeared sufficient to handle the barn dust without cleaning being required. At the end of the growing cycle for the second batch of birds, an estimated 3±2 mm film had developed.

Table 7.2 shows the power consumption of the supply and exhaust fans as measured during periodic checks. The two measurements for run 4 show the current drawn before and after the exhaust fan grille was cleaned. The supply fan current was at the maximum recommended value of 0.95A for most of the test period, with the notable exceptions of runs 2 and 3. The reason for the lower values could have been leakage occurring and lessening the power requirements of the fan (section 7.1.2.). The total power consumption did not vary more than 5% from 400 W for the entire test period.

7.1.2 Air flow

The ideal heat exchanger process would involve no mass transfer between the two air streams. Table 7.3 shows the

TABLE 7.1: MEAN DUST CONCENTRATIONS (particles/mL).

Particle size (10^{-6}m)	Run1	Run2	Run3	Run4
0.5 to 1.0	0.66	1.1	1.0	1.40
1.0 to 2.0	0.44	0.63	0.66	1.36
2.0 to 5.0	0.28	0.47	0.66	2.40
5.0 to 10.0	0.03	0.05	0.10	0.5
>10.0	0.004	0.01	0.01	0.03

TABLE 7.2: ELECTRICAL CONSUMPTION.

Run	Current to fan (A) Exhaust	Current to fan (A) Supply	Total power (W)
1	0.92	0.95	411
2	0.88	0.86	382
3	0.91	0.89	396
4a	0.87	0.93	396
b	0.83	0.93	387
5	0.89	0.95	405
6	0.87	0.95	400
7	0.85	0.94	393
8	0.86	0.95	398
9	0.88	0.95	403

a - before fan grille cleaned
b - after fan grille cleaned

increase in moisture content of the supply air for the nine individual runs. Differences of less than 1°C in the dewpoints could be accounted for by the measurement resolution and should be considered not significant. Runs 1, 2, and 3 were performed prior to December 19, 1985 which was the date the supply fan was relocated. Runs 4 through 9 were performed after the modification. The increase in humidity would indicate that leakage was occurring from either the exhaust side to the supply side or directly from the barn to the supply side. The laboratory test indicated that using negative pressure to draw the air through the heat exchanger

TABLE 7.3: DEWPOINT RISE IN SUPPLY AIR

Run	Change in dewpoint (°C)	Run	Change in dewpoint (°C)
1	13.0±1.8	6	0.07±0.4
2	13.6±0.9	7	-0.25±1.5
3	4.2±0.5	8	-0.64±0.8
4	0.05±0.2	9	0.05±0.07
5	-0.35±0.1		

TABLE 7.4: LEAKAGE RATES OF BARN AIR INTO SUPPLY SIDE OF HEAT EXCHANGER.

Run	Approximate Humidity (g/kg dry air)			Indicated leakage (% of total supply flow)
	Barn	Outside	Supply	
1	6.6	0.2	4.0	65
2	8.2	0.4	5.0	60
3	6.6	3.8	5.0	45
4	9.4	5.0	5.0	<5
5	5.0	2.0	2.0	<5
6	6.2	4.4	4.4	<5
7	8.8	0.2	0.2	<5
8	9.5	0.6	0.6	<5
9	7.8	4.4	4.4	<5

could result in excessive leakage rates and this was confirmed in the field. If the assumption is made that the air leaking into the supply air stream had a moisture content equal to the barn air, an estimation of the leakage could be calculated. Table 7.4 shows the estimated leakage rates that were occurring when the above assumption was made.

Another indicator of leakage was the temperature profile of the exhaust and supply air flows as they travel through the heat exchanger core. For the case of no leakage, the apparent heat lost by the exhaust air would equal the apparent heat gained by the supply air and would be the true heat transferred. If leakage was occurring between the two streams or between one stream and the outside air, the apparent heat transfer would be different for the two air streams. A comparison of the difference between the apparent heat transfer rates for each run appears in Table 7.5. The estimated leakage rates were calculated from the mean difference for each run. As with Table 7.4, the results strongly suggest that high leakage rates were occurring during runs 1, 2, and 3. Using the standard t-test (Harnett, 1982) differences in apparent heat transfer were also significantly different from zero for run 8. Assuming an accuracy of 0.5°C for any single thermistor (conservative), a difference of 20% could be accounted for by the calculation method. The estimated leakage rate for run 8 was significant (250 L/s) although the sign of the difference is opposite to that of the first three runs indicating leakage from the supply or outdoor air into the exhaust air. Run 8 was conducted before the problem of the missing styrofoam was noticed on February 24, 1986. Cold outdoor air could have leaked into the exhaust stream, causing the noticed difference in apparent heat transfer rates.

The measured air flow rates would be of little value for those runs for which leakage was a significant problem. The exhaust and supply flow rates for the remaining runs are shown in Table 7.6. The two sets of measurements for December 19, 1985 are for the cases before and after the exhaust fan grille was cleaned. Once the supply air was redirected upwards by means of the rectangular duct

TABLE 7.5: COMPARISON OF APPARENT HEAT TRANSFERRED FOR EACH SIDE OF HEAT EXCHANGER CORE.

Run	Enthalpy difference'	significant ($a_{0.05}$)
1	4.0±0.9	yes
2	3.6±1.1	yes
3	4.1±2.1	yes
4	0.2±0.2	no
5	-0.1±0.3	no
6	-0.2±0.7	no
7	-0.1±0.2	no
8	-0.5±0.2	yes
9	0.1±0.2	no

' - ($\Delta h_{supply} - \Delta h_{exhaust}) / \Delta h_{exhaust}$.

TABLE 7.6: MEASURED FLOW RATES.

Run	Flow rate (L/s) supply	exhaust
4(a)	590	360
(b)	590	400
5	545	495
6	540	500
7	585	410
9	n.m.	390

a - before exhaust fan grille cleaned
b - after exhaust fan grille cleaned
n.m. - not measured

(February 19, 1986), measurement of its flow was not possible. The flow rates were much lower than the 800 L/s predicted by using the manufacturer's performance data. Runs 5 and 6 were performed while the birds were 18 and 22 days old, while runs 4, 7, and 9 were performed while the birds were 40, 39, and 44 days old, respectively. The older birds required more ventilation and the resulting decrease in barn pressure increased the supply air flow rate and decreased the exhaust flow rate, as is reflected in Table 7.6.

7.1.3 Temperature recovery ratio

The average effectiveness (as calculated by equations 1.7 and 1.8) and the average TRR for each valid run are presented in Table 7.7. Although unacceptable leakage was shown to occur for run 8, the TRR was calculated since no leakage was occurring from the barn air to the supply air and therefore the temperature rise of the supply air accurately reflects the heat recovered for that run. Runs 4 and 7 were cases where the ambient temperature and barn humidity level were favorable for condensation to occur, making the supply air stream lower in apparent specific heat. In such cases, equation 2.7 is used to calculate effectiveness, and since the exhaust air flow rate was lower than the supply flow rate, equation 2.10 (for the calculation of TRR) reduces to the same form and the two values are equal. Runs 5, 6, and 9 were cases where the condensation present was not sufficient to make the apparent specific heat of the exhaust greater than the supply and, therefore, equation 2.8 was used for the effectiveness. The TRR for those cases is considerably lower than the effectiveness.

With the exception of run 8, the effectiveness varied little between all the runs with a high of 0.40 and a low of 0.34. This would be expected since effectiveness is a measure of how well the heat exchanger performs compared to an infinitely long counter-flow heat exchanger with the same flow parameters.

TABLE 7.7: MEASURED PERFORMANCE OF HEAT EXCHANGER

Run	Effectiveness	TRR
4	0.39	0.39
5	0.39	0.28
6	0.40	0.26
7	0.34	0.34
8	0.28	0.28
9	0.38	0.29

The temperature recovery ratio, on the other hand, reflects the true amount of heat being transfered to the supply air. The values of less than 0.3 for runs 5, 6, and 9 readily show a low rate of heat recovery. In these cases the low values resulted in part from the low exhaust flow rates produced by the Siemens fan. For the case at hand the high ventilation rates produced by the permanently-installed fans, as required for the broiler operation, reduced the heat recovery achieved by the heat exchanger, although the calculation of effectiveness shows no problem.

Figures 7.2 through 7.6 show how the TRR varied with some of the parameters measured during the various runs. To properly conclude how TRR would be affected by altering any one parameter would be extremely difficult based on these diagrams. The high barn humidities can be the result of high outdoor humidities, low barn ventilation rates, or high barn temperatures, all four of which could directly affect the TRR. The later runs would have the greatest amount of dust build-up on the plates, and dust could increase the TRR by decreasing the air flow rate; decrease the TRR by decreasing the heat transfer coefficient by insulating the plates; increase the TRR by increasing the heat transfer coefficient

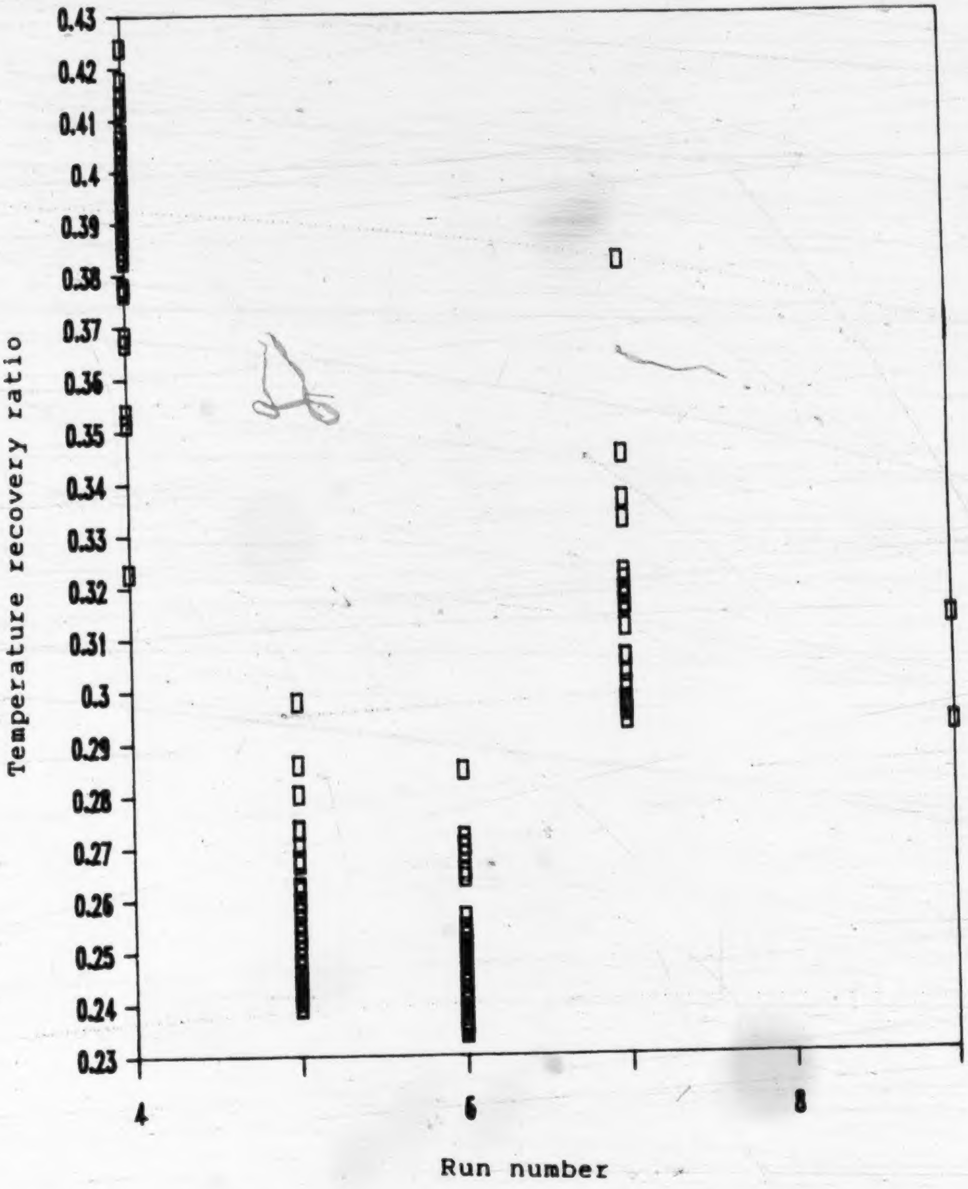


Figure 7.2 Variation of TRR with Run

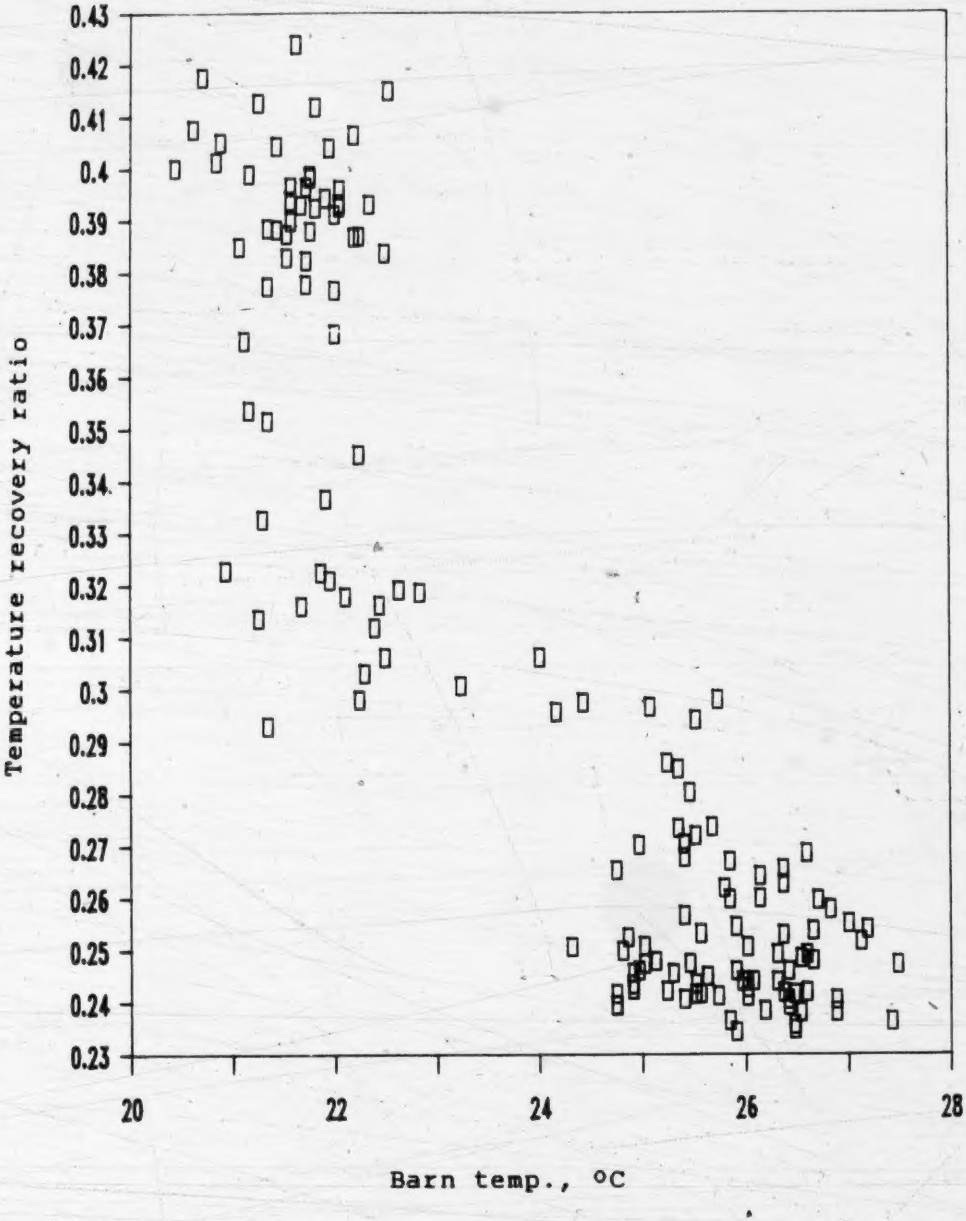


Figure 7.3 Variation of TRR with Barn temperature

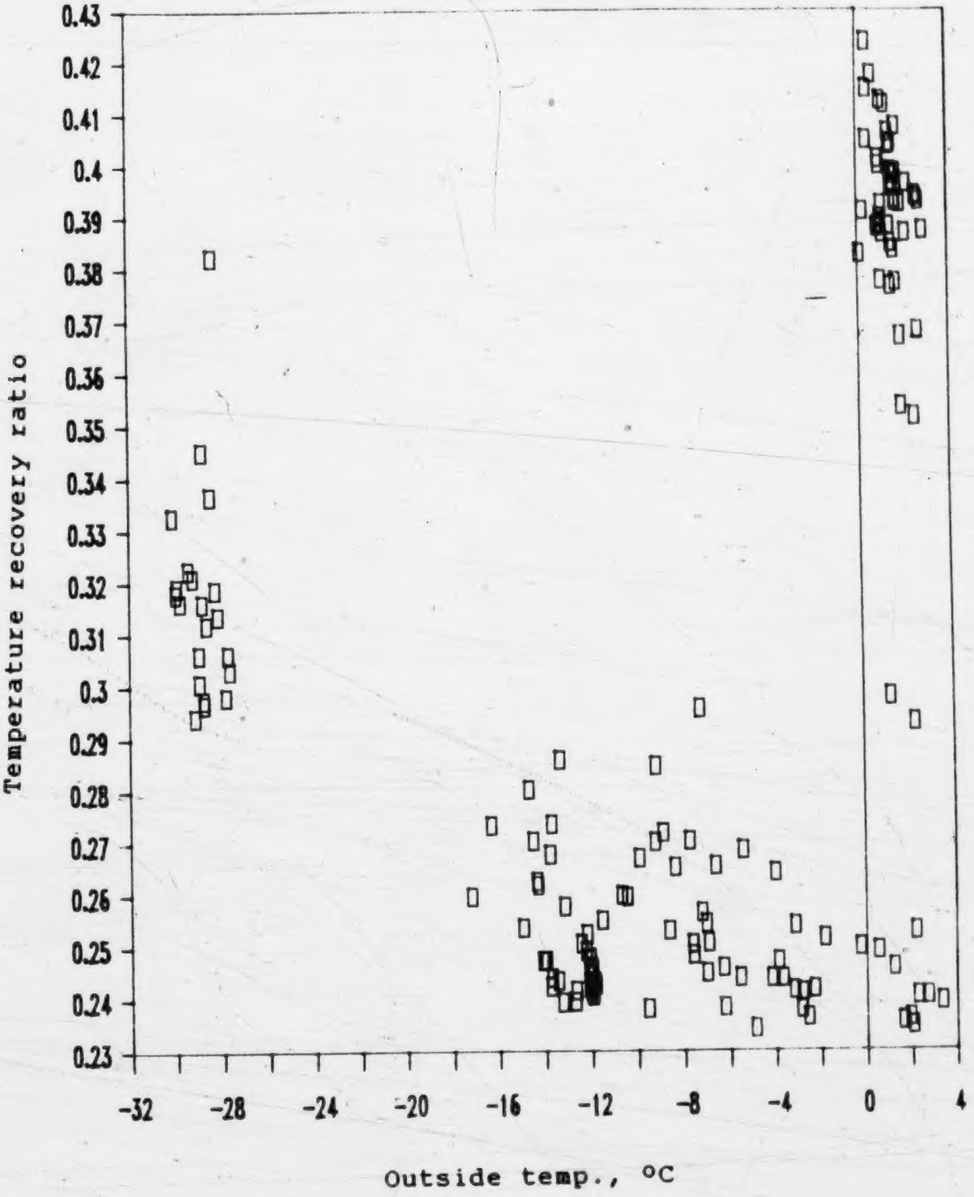


Figure 7.4 Variation of TRR with outside temperature

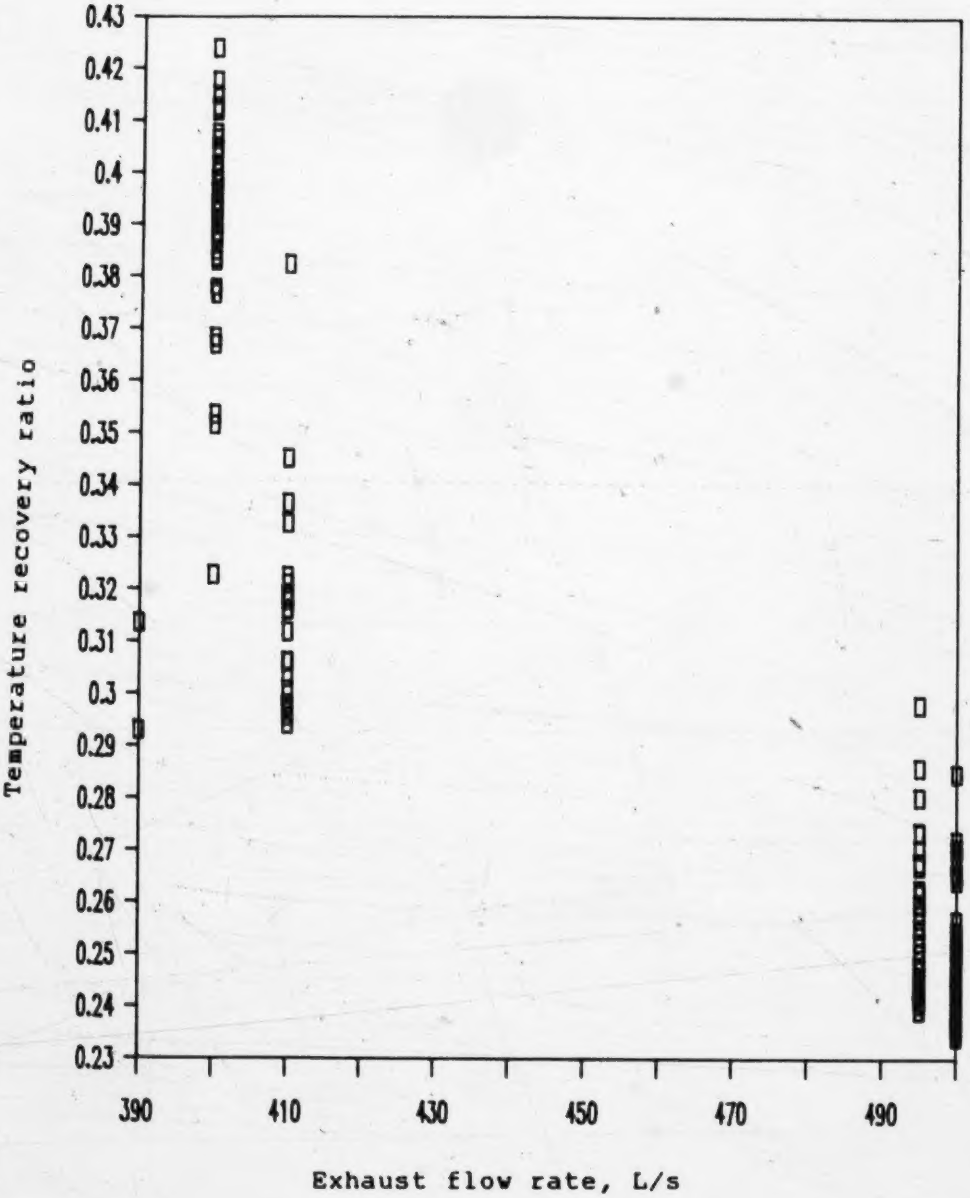


Figure 7.5 Variation of TRR with exhaust flow rate

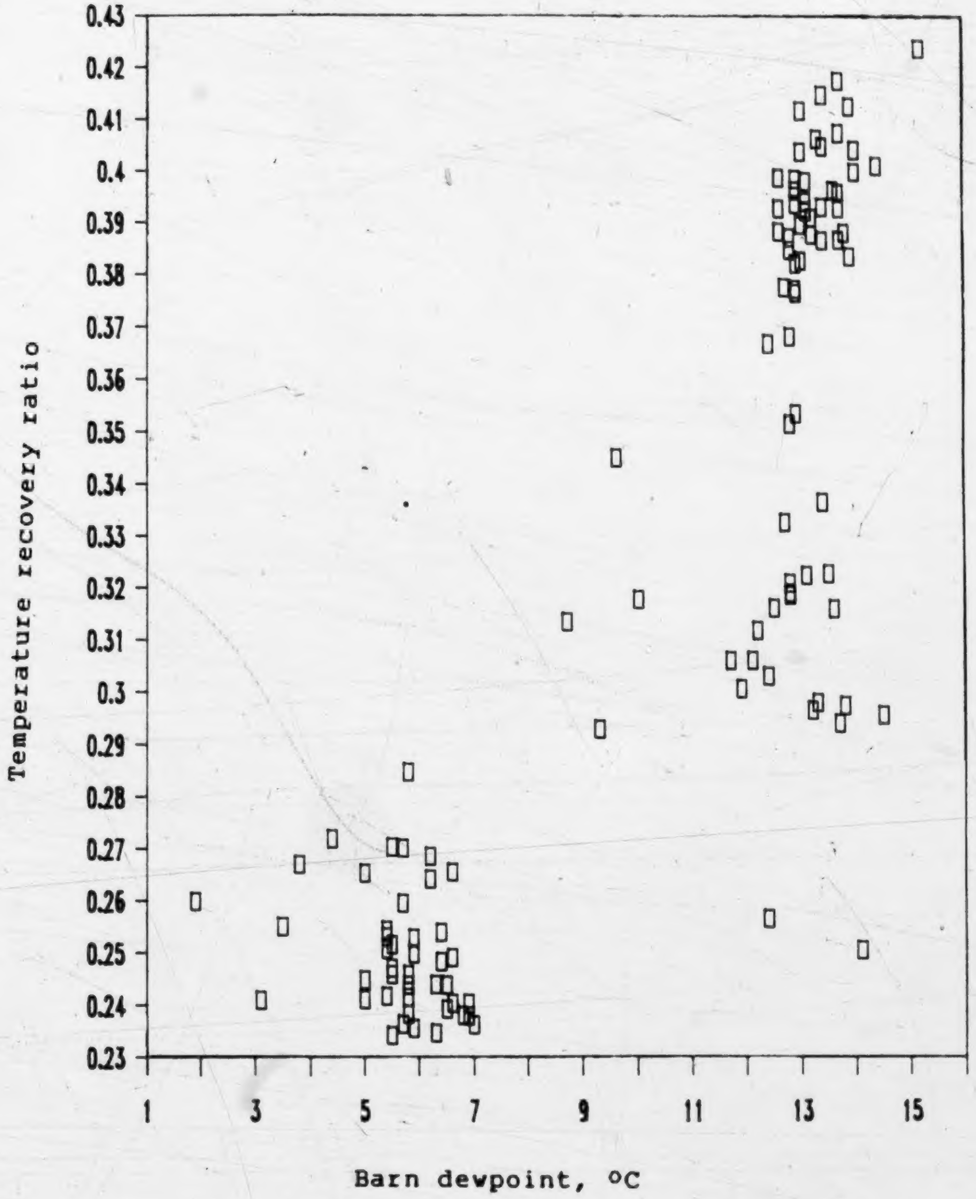


Figure 7.6 Variation of TRR with Barn dewpoint

by raising the air velocity inside the heat exchanger; or do all three at once. In order to better understand the factors involved it was deemed necessary to examine the local heat transfer coefficient along the heat exchanger core.

7.1.4 Heat transfer coefficient.

The local heat transfer coefficients were determined through the use of program HEAT, listed in Appendix E. The heat exchanger core was divided into three regions as determined by the positioning of the thermistors in the exhaust and supply air streams, as shown in Figure 6.5. Because the heat exchanger was in parallel flow, Region 1 represented the conditions at or near the inlets, Region 2 represented the middle conditions, and Region 3 represented the conditions near the outlets. The heat transfered was determined by noting the rise in supply air temperature, since no phase change was occurring on that side of the exchanger and the specific heat of the air stream should have been constant.

The heat transfer coefficients, as determined by the program HEAT, are listed in Table 7.8. Surprisingly, the region with the highest heat transfer coefficient was at the inlets (region 1) for all cases. The lowest heat transfer coefficient was in the center region (region 2) for all cases except run 7. Assuming an average air velocity of 2 m/s from the measured flow rates, the expected heat transfer coefficient from equation 3.1 would be 15 W/(m²°K). This is

TABLE 7.8: LOCAL HEAT TRANSFER COEFFICIENT BY POSITION IN HEAT EXCHANGER ($W/m^2 \cdot K$).

Run	Region 1	Region 2	Region 3
4	36.8±1.7	15.7±2.8	23.6±2.6
5	23.9±1.3	11.1±1.0	17.8±1.0
6	24.0±1.2	12.1±0.9	19.4±0.7
7	32.5±1.7	22.4±3.7	17.6±1.6
9	34.3±1.9	1.9±0.3	21.0±1.0
Ave	29.4±6.3	14.3±4.6	20.2±3.1

mean value ± standard deviation

TABLE 7.9: APPROXIMATE PORTION OF TOTAL HEAT TRANSFERRED BY REGION (%).

Run	Region 1	Region 2	Region 3
4	61	18	21
5	60	18	22
6	60	18	22
7	57	27	16
9	74	3	23

similar to the average value for region 2, but much lower than that for the other regions. The low values for the heat transfer coefficient in region 2 for run 9 are difficult to explain and the small amount of data collected for that run is not sufficient to draw a well-supported conclusion. Table 7.9 compares the actual portion of total heat transfered for each region and the approximate values adjusting for variations in heat transfer area between the regions. Over half of the total heat transfer occurred in the first one third of the heat exchanger.

The factors which were considered possible causes for variations in the heat transfer coefficient are as follows:

i) evaporation. As the exhaust air entered the heat exchanger core, it would not yet be saturated and it would be in contact with the condensate which was running down the plates.

ii) dust. There was no noticeable variation in dust build-up along the length of the heat transfer plates, although more dust was present on the plates for the later tests.

iii) condensation. The condensation of the barn moisture as the exhaust air cools should affect the heat transfer rate.

iv) flow rate. The variations in flow rate as measured could have caused changes in the heat transfer coefficients between the various runs.

In order to test the effects of the above factors on the heat transfer coefficients, multiple-linear regression models were set up for each of the three regions. The dependent variable in each case was the heat transfer coefficient, U in W/(m^2°K), and the independent variables used were as follows:

i) barn temperature, Tin (°C). Barn temperature would be negatively correlated with the onset of condensation, and would be positively correlated with evaporation.

ii) ambient temperature, Tout (°C). Ambient temperature would be negatively correlated with the onset of condensation.

iii) Reynolds number of the exhaust air stream, Nr (dimensionless). Since the exhaust air flow rate varied more than the supply flow rate, the former was used as a measure of the flow changes.

iv) relative humidity, Rh (as a fraction of unity). The relative humidity of the barn air should be positively correlated with the onset of condensation and be negatively correlated with evaporation.

v) date of run, D (in days). The number of days the unit was running should be positively correlated with the amount of dust build-up.

The multiple-linear regressions were accomplished through the use of the MINITAB computer language, as available on the MTS system at the University of Alberta. The five independent variables were matched with each of the dependent variables and then placed in columns to be read by MINITAB. The insignificant variables as determined by MINITAB could then be removed from the regression and the process continued.

The significant variables for the heat transfer coefficient in each region and the resulting regression equations appear in Table 7.10. The most interesting observation to be made from the results is that the high heat transfer coefficient present in region 1 was not significantly affected by relative humidity, barn temperature, or ambient temperature. The high correlation coefficient of 94.6% shows a very good agreement between the measured data and the regression equation produced. The

TABLE 7.10: LINEAR REGRESSION OF THE LOCAL HEAT TRANSFER COEFFICIENTS

REGION ONE:

$U1 = 85.7 - 0.00696\ Nr - 0.119\ t$ $r^2=94.6\%$

Analysis

Significant variable	Mean	Coefficient
Constant	30	85.7±1.2
Nr (Reynolds no.)	7100	-0.00696±0.00019
t (time, days)	20	-0.119±0.010

REGION TWO:

$U2 = 16.1 + 11.4\ Rh - 0.422\ T_{out} - 0.151\ t$

$r^2=71.2\%$

Analysis

Significant variable	Mean	Coefficient
Constant	14	16.1±2.7
Rh (relative humidity)	0.45	11.4±2.3
T_{out} (outside temperature,°C)	-14	-0.422±0.046
t (time in days)	20	-0.154±0.038

REGION THREE:

$U3 = 39.5 - 0.546\ T_{in} + 0.0755\ T_{out} - 0.0916\ t$

$r^2=71.4\%$

Analysis

Significant variable	Mean	Coefficient
Constant	20	39.5±1.8
T_{in} (barn temperature,°C)	-14	-0.546±0.098
T_{out} (outside temperature,°C)	-14	-0.0755±0.029
t (time, days)	20	-0.0916±0.023

factors affecting the heat transfer coefficient in this region appear to be the flow rate and the dust build-up. The negative correlation with Reynolds number can be explained as resulting from using the exhaust flow as the indicator of flow variation. If the supply flow Reynolds number had been used, a positive correlation would have resulted. It would appear that the heat transfer from the plate to the supply air was the limiting step in the heat transfer process.

The factors affecting the heat transfer coefficient in region 2 appear to be condensation and dust build-up. The value of U2 had a positive correlation with relative humidity and a negative correlation with ambient temperature, both of which indicate that the sooner the condensation begins in the heat exchanger, the higher the heat transfer coefficient in region 2.

The factors affecting the heat transfer coefficient in region 3 appear to be condensation and dust build-up as with region 2. The negative correlation with barn temperature indicates an increase in U3 with the early onset of condensation but the positive correlation with ambient temperature indicates the opposite. This is not readily explained, although if considerable moisture had condensed out in region 2, there would be a lower rate of condensation in region 3 and perhaps a lower heat transfer coefficient as well.

For all three heat exchanger regions, there was a significant negative correlation between heat transfer coefficient and the dust build-up. In all three regions the decrease amounted to approximately 0.1 W/(m^2°K) per day of operation. This was not expected based on the reported results from previous work (Stauffer and Vaughan, 1981). It would be reasonable to assume that the decrease would level out after a maximum dust thickness was reached.

The evidence given from the above multiple-linear regressions would suggest that the important processes involved in the heat exchanger are as follows:

i) flow development. For flow in enclosed channels, there is a length required for the fluid to travel before the flow becomes fully developed. (Holman, 1972). Generally, this starting length is not assumed to be significant when calculating heat transfer over the entire length of the transfer surface. The starting length required for the heat transfer properties to become fully developed is usually longer than that for the velocity profile and the former may take as much as 50 to 100 diameters to become totally settled. The thin boundary layers at the entrance would result in higher heat transfer coefficients in that region. The effect would be most readily observed for parallel-flow heat exchangers since the two inlets are at the same end. For counter-flow heat exchangers, the entrances are at opposite ends and the increased heat transfer coefficient would not be as readily noticed and, for the supply entrance, may be mistaken for an increase due to condensation on the exhaust side. Although this effect appears to be the most important as indicated by the very high heat transfer coefficients in region 1 of the heat exchanger, it was mostly overlooked by the previous studies. this effect was possibly responsible for the increase noted by Ogilvie (1967) for a parallel-flow laboratory test and attributed to the induced turbulence of the fans (section 1.6.1.).

ii) condensation. If the flow could be assumed to be nearly fully developed in region 2 as indicated by the drop in heat transfer coefficient from region 1 to region 2, any increase in the value of U from region 2 to region 3 could be attributed to the onset of condensation in the latter. This can be seen as the 50% average increase in the value of U (Table 7.8) from region 2 to region 3. This explanation is supported by the 25% increase in the average heat transfer coefficient noted by Giese and Bond (1952) during laboratory tests (section 1.6.1.). Since the portion of heat transferred in region 3 was only from 19 to 24% of the total, condensation can be assumed to have increased the total heat transferred by only 10%.

iii) dust. The presence of dust noticably decreases the heat transfer, presumably by decreasing the effective conductance of the medium seperating the two air streams. As indicated by the regressions, the measured drop in heat transfer coefficient was approximately 4

W/(m^2°K) in all three regions for the period from run 4 to run 9.

7.2 Economic feasibility

7.2.1 Capital cost

Table 7.11 lists the costs for fabricating a heat exchanger of the type designed and tested. A conservative estimate for the life of the device would be five years, which is a suitable period for taking a straight-line depreciation. The annual cost of the initial heat exchanger installation expense would be $230 per year.

7.2.2 Power cost

As determined in section 7.1.1. the power consumption of the fans was 400±20 W. A broiler barn would require heat for approximately 250 days a year. The heat exchanger operated for a total of 53 days for the two growing cycles (18 weeks). The expected operation time for one year, therefore, could be extrapolated to 115 days. Assuming an energy cost of $15/GJ, the total annual power cost for the heat exchanger would be $59.

7.2.3 Energy savings

The calculation of the reclaimed energy in a broiler barn involves several steps corresponding to the age of the birds. Table 7.12 shows the quantities relevant to

TABLE 7.11: FABRICATION COSTS.

Materials		
PVC	330	
plywood	40	
insulation	30	
fans	520	
controls and wiring	40	
misc. (glues, boards, etc.)	50	
Total materials		1010
Labor $8/hr.		
fabrication of core - 21 man-hours	168	
fabrication of housing - 4 man-hours	32	
installation - 1 man-hour	8	
Total labor cost		208
Total cost		1218

calculating a heat balance for the barn used, as measured by Feddes et al. (1984). The heat lost through the building shell, as measured in the above test, ranged from 1.6 to 2.2 MJ/h·K. The rounded value of 2.0 MJ/h·K was used in the following calculations. By doing a heat balance for each week, the heating deficit temperature can be determined from equation 3.5. The recoverable heat then can be found by using equation 3.8 and Tables 3.2 and 3.3 for each of the 9 weeks of the cycle and averaging the results. A summary of the calculations appears in Table 7.13. The prototype could be expected to save 94 GJ per year installed in the broiler barn.

TABLE 7.12: DATA[1] REQUIRED FOR ENERGY CALCULATIONS.

Week	Temp (°C)	Rh (%)	Vent. rate (L/s/1000 birds)	Heat produced by birds (kJ/hr/1000 birds)	Latent heat (kJ/hr/1000 birds)
1	33	30	(fans turned off)		
2	28	41	92	7.9	4.8
3	27	38	170	12.6	7.9
4	24	43	304	23.4	12.2
5	23	41	382	40.3	18.6
6	22	58	301	43.2	19.6
7	21	48	575	57.2	27.2
8			(barn empty)		
9			(barn empty)		

[1] Feddes et al. (1984)

TABLE 7.13: ANNUAL EXPECTED HEAT RECOVERY OF PROTOTYPE.

Week	Deficit temp. °C	Heat exchanger	Heat recovery (GJ/year)
1	30	off	0
2	23	on 2 days	5
3	22	on	21
4	17	on	18
5	12	on	17
6	7	on	16
7	10	on	17
8	-	off	0
9	-	off	0
Total			94

TABLE 7.14: ECONOMIC ANALYSIS OF THE PROTOTYPE

	Fuel type			
	Coal	Natural gas	Propane	Elect.
Approx. cost ($/GJ)	0.70	4.00	10.00	15.00
Annual heat savings ($)	66	376	940	1410
Annual elect. cost ($)	45	45	45	45
Depreciation (20%)	245	245	245	245
Net return ($)	(loss)	86	650	1120
Rate of return on invest (%)	(loss)	7.6	57	97

7.2.4 Rate of return on investment

Table 7.14 shows the expected rate of return on the original $1140 invested in installing the heat exchanger. The unit was not economical in the barn in which it was installed because the low energy cost of coal made the value of the energy recovered approximately equal to the cost of electricity to run the fans. The prototype was marginally economical for heating with natural gas and very feasible for higher cost energy sources. The lower fan flow rates than those for which it was designed significantly affected the heat recovery. If the flow rates had been 800 L/s instead of those measured, the resulting heat savings would have been approximately 30% higher, giving a rate of return of 18% for the investment. For regions colder than Edmonton (Grande Prairie, for example) the rate of return would be logically higher, whereas for warmer regions, (Lethbridge) the heat exchanger would be less economical.

8. SUMMARY AND CONCLUSIONS

8.1 Describing heat exchanger performance

The measure of a heat exchanger's performance should reflect the amount of heat recovered. The method devised in this thesis for describing heat exchanger performance is the temperature recovery ratio, TRR, the method of calculation for which is given in equation 2.10.

$$TRR = \frac{M_c * (T_{co} - T_{ci})}{M_{max} * (T_{wi} - T_{ci})} \qquad 2.10$$

8.1.1 Temperature recovery ratio

For most agricultural applications, the most desirable TRR would be in the range of 0.4 to 0.6, as determined by the return on increasing the heat transfer area. Increasing the area was determined to become uneconomical when the slight increase in energy savings does not warrant the extra capital cost. For many applications (such as swine housing) a TRR in the range given above can maintain the optimum barn temperature without supplemental heating. A low TRR may not recover sufficient heat to be economically viable.

8.1.2 Flow direction

Cross-flow heat exchangers should not be used due to the large variations in temperature across the heat transfer plates making ice formation possible even though the bulk temperature of the exhaust is above freezing. Parallel-flow heat exchangers can perform satisfactorily in barns, as shown by the unit developed and tested for this thesis. The choice between parallel- and counter-flow should be made on the basis of ease of installation for the particular application.

8.1.3 Heat transfer coefficient

The heat transfer coefficients measured within the test unit ranged from 11.1 to 36.8 W/(m^2°K). For the parallel-flow heat exchanger tested, the region with the highest heat transfer coefficient was near the inlets while the lowest heat transfer coefficient was located in the central region. The entrance effects produced before the flow becomes fully developed appear to have the greatest effect on heat transfer. Condensation also appeared to significantly increase the heat transfer coefficient. For the unit tested, condensation occurred near the outlets where the temperature difference of the two air streams was low and the total heat transferred was increased about 10% due to condensation. For counter-flow, the heat transferred should be affected less by the entrance effects and more by condensation than for parallel-flow.

8.1.4 Dust

Dust was found to decrease the heat transfer coefficients by about 4 W/m^2·K over a 40 day period.

Plugging of the heat exchanger with dust was not encountered although an estimated 4mm thick film had developed by the end of the test period. A spacing of 15mm or greater between the heat transfer plates should be sufficient to prevent plugging.

8.1.5 Condensation

The literature review suggested the exhaust air stream should flow vertically upward causing the condensate to flow down the heat transfer plate, aiding in the prevention of fouling by dust. A 50mm diameter hole was found to be sufficient to remove the condensate from the heat exchanger. In some applications the condensate could be channelled to an already existing drain but the addition of a drain hole was necessary for the test site.

8.1.6 Geometry

Flat-plate heat exchangers are the most practical and economical of the various types of heat exchangers. The polyethylene film and multiple flat-plate designs were shown to have similar operating costs. In studying the heat transfer coefficient of the heat exchanger built, entrance effects increased the heat transfer at the inlets. The typically long flow channels (20 m) of the polyethylene film designs would not take advantage of the entance effects to as great a degree as would the shorter channels of the multiple flat-plate designs. The portability and compactness of the latter are also definite advantages.

8.1.7 Fan selection

Fan selection was found to greatly influence the performance and feasibility of a heat exchanger. Assuming equal maintenance costs, the less expensive fans with low efficiencies were generally determined to be more economical than higher priced, more efficient fans.

8.1.8 Economic feasibility

The conditions which would favor an acceptable rate of return for heat exchange are high fuel costs, high ventilation rates, cold ambient temperatures, and low capital costs. For the broiler barn tested, a capital cost of approximately $2 per L/s of supply air for the heat exchanger was marginally acceptable using natural gas as a fuel source, and not acceptable using coal.

9. RECOMMENDATIONS

9.1 Modifications

The highest heat transfer coefficients were found to be nearest the inlets for parallel-flow heat exchangers which is also the region of highest temperature difference. Less heat transfer plate area should be required for equal performance if more of the heat transfer was to take place near the inlets. A more favorable design than the one used would be one with a higher number of shorter heat transfer plates. To accomplish this while not changing the air velocity, the plates would need to be closer together. From the test results, a spacing of less than 15 mm would greatly increase the possibities for plugging by dust. An improvement on the prototype would be to use heat transfer plates with the dimensions of 610 x 1220 mm (four per PVC sheet) and space them at 15mm spacings.

9.2 Feasibility of heat exchangers

The physical problems involved with heat exchange can be overcome by a farm fabricated device. By keeping costs as low as possible, there are agricultural applications in which heat recovery through the use of heat exchangers can be economically viable. Increasing fuel costs and possible increases in recommended minimum ventilation rates would make heat exchangers more attractive.

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```
*  PARALLEL-FLOW EXCHANGER SIMULATION
*  PROGRAMMING LANGUAGE: CSMP
*  The following macro calculates the specific heat for air
*         TDB- dry bulb temperature (c)
*         TSAT - saturation temperature
*         W - humidity ratio

MACRO ENTHL=SPECH(TDB,TSAT,W)
     PROCEDURAL
 IF (TSAT .GT. TDB) GOTO 99
     ENTHL = 1006+1840*W
     GOTO 100
99 ENTHL=NLFGEN(DENTH,TDB)
100 CONTINUE
ENDMAC
INITIAL

* The following parameters are used:
*        TIN - inside temperature (c)
*        TOUT - outside temperature (c)
*        U - overall heat transfer coefficient (W/K/m**2)
*        HEIGHT - height of heat exchange surface (m)
*        TSAT -  dew point of inside air (c)

  PARAMETER  TIN=20,TOUT=-40,U=25,HEIGHT=0.7,TSAT=20.

* WRAT is the function representing humidity ratio.

  FUNCTION WRAT=(-55,.0000129),(-50,.0000243),...
     (-45,.0000445),(-40,.00008),(-30,.00023),...
     (-20,.00064),(-10,.0016),(-5,.002708),(0,.00379),...
     (5,.00542),(10,.00766),(15,.0107),(20,.0148),...
     (25,.0202),(30,.0276),(35,.0368)

*  DENTH is the function representing enthalpy change per
*  degree for air following the saturation curve, using
*  temperature as the independant variable.

 FUNCTION DENTH=(-4.5,1564),(.5,1730),(5.5,2005),...
      (10.5,2372),(15.5,2850),(20.5,3480),(25.5,4298),...
      (30.5,5363),(35.5,6754)

*  Calculate the initial humidity ratios for each air stream

  W1=NLFGEN(WRAT,TSAT)
  W2=NLFGEN(WRAT,TOUT)

*  VEL - the velocity of the air streams (m/s)
*  MDOT - the mass flow rate (kg/s)
*  SPEC2 - the specific heat of the supply stream
```

```
VEL = 4.0
MDOT = 1.25

* The heat flow is determined by taking a time based
* derivative of the incremental heat transfer and then
* integrating using the INTGRL command since direct
* summation is difficult in the dynamic section of CSMP.
* The heat flow derivative is therefore approximated by:
*     QD = (U*HEIGHT*LENGTH*Dtemp) / time
* where Dtemp is the local temperature difference.
* The total heat transferred, QTOTAL is the integral of QD.
* LENGTH/time is the velocity, VEL.

UPRIME = U * HEIGHT * VEL
*
*   SPEC1 - the local specific heat of the exhaust air
*   SPEC2 - the specific heat of the supply air (assumed
*                                               constant)
 SPEC2 = SPECH(TOUT,-50,W2)
DYNAMIC
*
* T1 - local temperature of the exhaust stream
* T2 - local temperature of the supply stream
* TMEAN - log mean temperature difference
* ERROR - difference between program and predicted length
*
 DTEMP=T1-T2
 QD = (DTEMP) * UPRIME
 SPEC1 = SPECH(T1,TSAT,W1)
 T1D = -QD/MDOT/SPEC1
 T2D = QD/MDOT/SPEC2
 T1 = INTGRL(TIN,T1D)
 T2 = INTGRL(TOUT,T2D)
 LENGTH = VEL * TIME
 QTOTAL = INTGRL(0.0,QD)
 TRAT = (TIN - TOUT) /DTEMP
  IF (TRAT .LE. 1.1) GOTO 1      ' prevent division by zero
  TMEAN = (TIN-TOUT-DTEMP)/ALOG(TRAT)
  PRLEN = QTOTAL/(U*HEIGHT*TMEAN)
  ERROR = (PRLEN-LENGTH)/PRLEN*100
 1 CONTINUE
*
TERMINAL
 METHOD RKSFX
 TIMER FINTIM = 100,DELT=1, PRDEL=10
 FINISH T1=0.0,DTEMP=2.0
 PRINT T1, T2, DTEMP, LENGTH, PRLEN, ERROR
END
STOP
ENDJOB
```

```
*   .COUNTER FLOW EXCHANGER SIMULATION
*  PROGRAMMING LANGUAGE: CSMP
*  The following macro is the same as in PARALLEL case
*         TDB- dry bulb temperature (c)
*         TSAT - saturation temperature
*         W - humidity ratio

MACRO ENTHL=SPECH(TDB,TSAT,W)
     PROCEDURAL
 IF (TSAT .GT. TDB) GOTO 99
     ENTHL = 1006+1840*W
     GOTO 100
99 ENTHL=NLFGEN(DENTH,TDB)
100 CONTINUE
ENDMAC

* This macro determines the enthalpy for the air given the
* same variable definitions as above. The derivative of the
* set of values obtained from this macro would be the same
* as the set of values determined above (the specific heat).

MACRO H=UVAL(TDB,TSAT,W)
 PROCEDURAL
  IF (TSAT .GT. TDB) GOTO 98
  H = 1006*TDB + W*(2.501E6 + 1840 * TDB)
  GOTO 101
98 H = NLFGEN (ENT,TDB)
101 CONTINUE
ENDMAC
INITIAL

* The following parameters are used:
*       TIN - inside temperature (c)
*       TOUT - outside temperature (c)
*       U - overall heat transfer coefficient (W/K/m**2)
*       HEIGHT - height of heat exchange surface (m)
*       TSAT -  dew point of inside air (c)

  PARAMETER  TIN=20,TOUT=-40,U=25,HEIGHT=0.7,TSAT=0,GO=0.0

* WRAT is the function representing humidity ratio for a
*      given temperature.

  FUNCTION WRAT=(-55,.0000129),(-50,.0000243),...
     (-45,.0000445),(-40,.00008),(-30,.00023),...
     (-20,.00064),(-10,.0016),(-5,.002708),(0,.00379),...
     (5,.00542),(10,.00766),(15,.0107),(20,.0148),...
     (25,.0202),(30,.0276),(35,.0368)

*  DENTH is the same as for the parallel case

 FUNCTION DENTH=(-4.5,1564),(.5,1730),(5.5,2005),...
      (10.5,2372),(15.5,2850),(20.5,3480),(25.5,4298),...
      (30.5,5363),(35.5,6754)
```

```
* ENT is the function for enthalpy of saturated air given
* the dry bulb temperature.

 FUNCTION ENT=(-40,-40040),(-30,-29600),(-20,-18550),(...
      -10,-6070),(-5,1164),(0,9473),(5,18640),(...
      10,29350),(15,42110),(20,57560),(25,76500),...
      (30,100010),(35,129450)

*  Calculate the initial humidity ratios for each air stream

  W1=NLFGEN(WRAT,TSAT)
  W2=NLFGEN(WRAT,TOUT)

*  VEL - the velocity of the air streams (m/s)
*  MDOT - the mass flow rate (kg/s)
*  SPEC2 - the specific heat of the supply stream

VEL = 4.0
MDOT = 1.25

* The heat flow is determined as with the parallel case

UPRIME = U * HEIGHT * VEL

*   FIRST - the initial enthalpy of the exhaust air
*   FINAL - the enthalpy of the exhaust air as it leaves
*           the system
*   DT2   - the total change in supply air temperature
*   SUPLY - the temperature of the supply air as it comes
*           into the barn
*   SPEC1 - the local specific heat of the exhaust air
*   SPEC2 - the specific heat of the supply air
*
 SPEC2 = SPECH(TOUT,-60,W2)
 FIRST = UVAL(TIN,TSAT,W1)
 FINAL = UVAL(GO,TSAT,W1)
*
* do first law of thermo on the two air streams to determine
* SUPLY
*
 DT2 = (FIRST - FINAL)/SPEC2
 SUPLY = TOUT + DT2
*
DYNAMIC
*
* T1 - local temperature of the exhaust stream
* T2 - local temperature of the supply stream
*
 QD = (T1 - T2) * UPRIME
 SPEC1 = SPECH(T1,TSAT,W1)
 T1D = -QD/MDOT/SPEC1
 T2D = -QD/MDOT/SPEC2
 T1 = INTGRL(TIN,T1D)
 T2 = INTGRL(SUPLY,T2D)
 LENGTH = VEL * TIME
 QTOTAL = INTGRL(0.0,QD)
 TRAT = (TIN - SUPLY)/DTEMP
  IF (TRAT .LE. 1.1) GOTO 1      ' prevent division by zero
  TMEAN = (TIN-SUPLY-DTEMP)/ALOG(TRAT)
  PRLEN = QTOTAL/(U*HEIGHT*TMEAN)
  ERROR = (PRLEN-LENGTH)/PRLEN*100
 1 CONTINUE
*
TERMINAL
 METHOD RKSFX
 TIMER FINTIM = 100,DELT=1, PRDEL=10
 FINISH T1=0.0,DTEMP=2.0
 PRINT T1, T2, DTEMP, LENGTH, PRLEN, ERROR
END
STOP
ENDJOB
```

12. APPENDIX B: COMPUTER PROGRAM - ECONOMIC COMPARISON OF GEOMETRIES

```
5  ' PROGRAMMING LANGUAGE: BASIC
10 ' This program determines the pressure drops and the
11 ' heat transfer area required for parallel plate and
12 ' shell-and-tube heat exchangers for a size of
13 ' 1.0 NTU. It then calculates the annual cost for
14 ' operating each based on the above (neglecting labor).

20 DIM COSTUB(16)        ' cost of tubing by diameter
25 DATA .5,.65,.8,1.2,1.7,2.2,2.3,2.4
30 DATA 2.5,2.8,3.2,3.5,4.2,5.,5.3,6.0
31 FOR I=1 TO 16
32 READ COSTUB(I)
33 NEXT
35 PLYCST=4.              ' cost of plywood in $/sq.m.
40 PLATIC=6.              ' cost of plastic plate
45 POLY=.31               ' cost of poly film
50 GAMMA=1.41E-5          ' kinematic viscosity of the air
55 EFFIC=.33              ' fan efficiency
60 PI=3.14
100 '  Shell-and-tube design
105 MINCOST=10000         ' initialize the minimum cost
110 FOR J=1 TO 10
120 FOR I=1 TO 16
125 D=I*.005              ' choose the tube diameter
130 V=J                   ' choose the velocity
135 TUBE=.8/(PI*D*D*V/4.) ' number of tubes
137 PER=TUBE**.5*D*10.    ' perimeter of the shell
140 Q=PI*D*D*V/4.         ' flow rate per tube
145 M=1.2*Q               ' mass flow
150 U=10.*V**.48          ' estimate heat transfer rate
155 A=1000.*M/U           ' area required for 1.0 NTU
160 L=A/(PI*D)            ' length of tube
165 RN=D*V/GAMMA          ' Reynolds number
170 GOSUB 1000
180 HL=(F*L/2./D+.75)*V*V ' head loss * g
190 PRESS=HL*1.2          ' pressure loss
200 AR=PER*L              ' plywood area
210 POWER=2.*PRESS*.8     ' fan power for both sides
214 ACOST = POWER*.234/EFFIC
215 ANCST=(AR*PLYCST+TUBE*L*COSTUB(I))*.25
216 ANCOST=ANCST+ACOST
217 REM  ANCOST now represents the annual cost of running
218 REM  the heat exchanger.
220 IF ANCOST > MINCOST THEN GOTO 240
225 MINCOST = ANCOST
230 VEL = V
235 SIZE = D
236 CPOW=ACOST
237 CMAT=ANCST
240 NEXT I
241 NEXT J
```

```
245 PRINT "SHELL-AND-TUBE   vel = " VEL " dia =" SIZE
250 PRINT  "MATERIALS" CMAT "POWER" CPOW "TOTAL" MINCOST
251 REM
300 REM  Multiple flat-plate
305 MINCOST=10000          ' reset minimum cost
310 FOR JJ=1 TO 10
315 FOR II=1 TO 16
316 W=II*.005              ' distance between plates
320 V=JJ                   ' velocity
325 Q=W*V/2                ' flow for a meter width of plate
330 M=1.2*Q                ' mass flow
335 U=10.*V**.48           ' estimate heat transfer
340 L=1000.*M/U            ' length required for 1 NTU
345 RN=2.*W*V/GAMMA        ' Reynolds number
350 GOSUB 1000
355 HL=(F*L/4/W +.75)*V*V  ' head loss * g
360 PRESS=HL*1.2           ' pressure loss
365 SPECA=L/Q              ' area of plate per flow
370 ARC=1.6/V              ' cross sectional area
375 PER=ARC**.5*4.         ' perimeter
380 AR=PER*L               ' outer area of exchanger
385 AREA=SPECA*.8          ' plate area required
390 POWER=2*PRESS*.8       ' fan power
395 ACOST = POWER*.234/EFFIC
400 ANCST=(AR*PLYCST+AREA*PLATIC)*.25
405 ANCOST=ANCST+ACOST
410 REM  ANCOST now represents the annual cost of running
415 REM  the heat exchanger.
420 IF ANCOST > MINCOST THEN GOTO 440
425 MINCOST = ANCOST
430 VEL = V
435 SIZE = W
436 CPOW=ACOST
437 CMAT=ANCST
440 NEXT II
441 NEXT JJ
445 PRINT "MULTIPLE PLATE   vel = " VEL " space=" SIZE
450 PRINT  "MATERIALS" CMAT "POWER" CPOW "TOTAL" MINCOST
600 REM polyfilm heat exchanger
605 MINCOST=10000          ' reset minimum cost
610 FOR K=1 TO 4           ' air velocity limited to 4m/s
615 FOR KI=1 TO 16
620 V=K
625 W=KI*.01               ' width of air passage
630 M=1.2*.8               ' mass flow rate
635 U=10.*V**.48           ' estimate heat transfer rate
640 AREA=1000.*M/U         ' poly area required
645 RN=2.*W*V/GAMMA        ' Reynolds number
650 GOSUB 1000
655 HL=(F*L/4/W +.75)*V*V  ' head loss * g
660 PRESS=HL*1.2           ' pressure loss
665 H=.8/W/V               ' width of poly
670 L=AREA/H               ' length of poly
675 AR=AREA*2+W*L*4        ' outer area of exchanger
```

```
690 POWER=2*PRESS*.8        ' fan power
695 ACOST = POWER*.234/EFFIC
700 ANCST=(AR*PLYCST+AREA*POLY)*.25
705 ANCOST=ANCST+ACOST
710 REM  ANCOST now represents the annual cost of running
715 REM  the heat exchanger.
720 IF ANCOST > MINCOST THEN GOTO 740
725 MINCOST = ANCOST
730 VEL = V
735 SIZE = W
736 CPOW=ACOST
737 CMAT=ANCST
740 NEXT KI
741 NEXT K
745 PRINT "POLY FILM  vel = " VEL " space=" SIZE
750 PRINT  "MATERIALS" CMAT "POWER" CPOW "TOTAL" MINCOST
755 BEEP                     ' signal termination
760 END
998 '   The following subroutine approximates a Moody
999 '   diagram.
1000  IF RN > 2000  THEN GOTO 1091
1010  F=64./RN               ' laminar flow
1020  RETURN
1091   IF RN > 3000 THEN GOTO 1192
1095  F=.03+1E-5*(RN-2000.)
1100  RETURN
1192   IF RN > 1E6 THEN GOTO 1293
1195   F=23900./(RN+600000.)
1200   RETURN
1293   F=.015
1300   RETURN
```

13. APPENDIX C: COMPUTER PROGRAM - WEATHER SIMULATION

```
*  PROGRAMMING LANGUAGE: FORTRAN
*
*  DEPARTMENT OF AGRICULTURAL ENGINEERING
*  UNIVERSITY OF ALBERTA
*  EDMONTON , ALBERTA
*
*  " DEGREE - DAYS  BELOW A VARIABLE BASE TEMPERATURE "
*
*  PROGRAMMED BY : ANDREAS AALHUS
*  DATE          : MAY 27 , 1985
***** UPDATED BY : DONALD KENNEDY
***** DATE       : DEC 4, 1985
*
*  PURPOSE : using easily available  daily temperature data,
*    it was desired to determine the approximate number of
*    degree days below a certain base temperature which is
*    inputted by the user.
***UPDATE : expanded to also output the time (in days) below
***   the base temperature.
*
*  THEORY : the program assumes that a day's minimum
*    temperature occurs at midnight and increases linearly
*    to the day's maximun which occurs at noon. The
*    temperature then decreases linearly to the next day's
*    minimum temperature. This pattern of lines approximates
*    the temperature fluctuations which occur over a period
*    of days.  Portions of the temperature lines will be
*    below the base temperature line.  The area between
*    these portions and the base temperature line will be an
*    approximation of the degree-days below that base.
*
*  THINGS TO KNOW : the program is an interactive one which
*    will request from and supply information to the user.
*    - the program is designed to handle data for up to 100
*    different locations .  Initially however, the data file
*    contains data for only 22 Alberta locations.
*    - the data file is arranged as follows :
*       1) number of locations
*       2) names of and location numbers
*       3) number of months for which data is available (12)
*       4) names of months
*       5) number of days in each month
*       6) daily maximum temperatures for each location
*       7) daily minimum temperatures for each location
*
* DECLARE THE VARIABLES AND /OR ARRAYS TO BE USED IN THE
* PROGRAM .
*
      INTEGER I,J,K,M,N,COL,CORLOC,CORT,DAY,DAYSM(12),LOCATN
     + ,NLOC,NMON,ROW
*
```

```
      REAL BASET,B1,B2,DHR(12),DHR1,DHR2,TMAX(365,100)
     +     ,TMIN(366,100),Y1M,Y2M,TIME(12),HOURS,HR1,HR2
*
      CHARACTER*30 LOC(100)
      CHARACTER*12 MON(12)
*
*
* WHERE THE VARIABLES ARE USED AS FOLLOWS :
*
* I,J,K,M,N     - used as counters in do-loops.
* COL,ROW         - used as counters in implied do-lists.
* CORLOC,CORT     - used to determine if the location and
*   base temperature entered by the user are the ones which
*   were intended ( CORLOC=correct location , CORT=correct
*   temperature ).
* DAY   -  represents the cumulative day of the year .
* DAYSM(12)       - an array which holds the number of days
*   in each month .
* LOCATN          - used to hold a number which indicates
*   the location being investigated .
* NLOC,NMON       - the number of locations and months for
*   which data is available .
* BASET           - the base temperature.
* B1,B2           - the y-intercepts of straight lines drawn
*   between the maximum and minimum temps for the day.
* DHR(12)         - an array which holds the number of
*   degree-hours for each month.
* DHR1,DHR2       - used to hold intermediate values.
* HR1,HR2         - the times at which the temperature
*                  drops below thw base temperature.
* TMAX(365,100)   - an array which holds the daily maximums.
* TMIN(366,100)   - an array which holds the daily minimums.
* Y1M,Y2M         - the slopes of lines between the
*                  maximum and minimum temperatures.
* LOC(100)        - an array which holds the location names.
* MON(12)         - an array which holds the month names.
* TIME(12)        - contains the amount of time in days
*      below the base for each month.
* HOURS           - used to sum the time below the base.
*
* INITIALISE THE FOLLOWING VARIABLES AND/OR ARRAYS TO THEIR
* STARTING VALUES .
*
*
      I=0
      J=0
      K=0
      M=0
      N=0
      COL=0
      ROW=0
      CORLOC=0
      CORT=0
      DAY=0
      LOCATN=0
      NLOC=0
      NMON=0
      BASET=0.0
      B1=0.0
      B2=0.0
      DHR1=0.0
      DHR2=0.0
      HR1=0.0
      HR2=0.0
      Y1M=0.0
      Y2M=0.0
      HOURS=0.
   50 DO 60 I=1,12
   51   DAYSM(I)=0
   52   DHR(I)=0.0
   53   MON(I)=' '
        TIME(I)=0.
   60 CONTINUE
*
   70 DO 80 I=1,100
   71   LOC(I)=' '
   80 CONTINUE
*
   90 DO 110 ROW=1,365
   91   DO 100 COL=1,100
   92     TMAX(ROW,COL)=0.0
  100   CONTINUE
  110 CONTINUE
*
  120 DO 140 ROW=1,366
  121   DO 130 COL=1,100
  122     TMIN(ROW,COL)=0.0
  130   CONTINUE
  140 CONTINUE
*
* READ IN THE NUMBER OF LOCATIONS FROM THE DATA FILE .
*
  145 READ(4,1) NLOC
    1 FORMAT(1X,I4)
*
* READ THE LOCATION NAMES INTO A CHARACTER ARRAY .
*
  150 READ(4,2) (LOC(I),I=1,NLOC)
    2 FORMAT(1X,A30)
*
* READ IN THE NUMBER OF MONTHS FOR WHICH DATA IS AVAILABLE.
*
  155 READ(4,3) NMON
    3 FORMAT(1X,I3)
*
* READ THE NAMES OF THE MONTHS INTO A CHARACTER ARRAY.
*
  160 READ(4,4) (MON(J),J=1,NMON)
```

```
    4 FORMAT(1X,A12)
*
* READ IN THE NUMBER OF DAYS IN EACH MONTH .
*
  165 READ(4,5) (DAYSM(K),K=1,NMON)
    5 FORMAT(1X,12I4)
*
* READ IN THE MAXIMUM DAILY TEMPERATURES .
*
  170 READ(4,23) ((TMAX(ROW,COL),COL=1,11),ROW=1,365)
      READ(4,23) ((TMAX(ROW,COL),COL=12,NLOC),ROW=1,365)
*
* READ IN THE MINIMUM DAILY TEMPERATURE .
*
  175 READ(4,23) ((TMIN(ROW,COL),COL=1,11),ROW=1,366)
      READ(4,23) ((TMIN(ROW,COL),COL=12,NLOC),ROW=1,366)
   23 FORMAT(1X,11F6.1)
*
* ASK THE USER FOR THE DESIRED LOCATION .
*
  180 WRITE(6,6)'KEY IN THE NUMBER OF THE LOCATION YOU ARE'
     +         ,' INTERESTED IN .  THE LOCATIONS ARE :'
    6 FORMAT(T2,A,/,T2,A)
*
* LET THE USER KNOW THE AVAILABLE LOCATIONS .
*
  185 WRITE(6,7) (LOC(I),I=1,NLOC)
    7 FORMAT(T2,A30)
*
* READ IN THE NUMBER OF THE DESIRED LOCATION .
*
  190 READ(5,*) LOCATN
*
* TEST TO SEE IF THE ENTERED LOCATION NUMBER IS VALID. IF IT
* IS NOT, THEN ASK FOR IT TO BE RE-ENTERED .
*
  200 IF((LOCATN.LT.1).OR.(LOCATN.GT.NLOC)) THEN
          WRITE(6,8)'THAT IS NOT A LOCATION NUMBER.'
     +              ,'PLEASE RE-ENTER A LOCATION .'
     +              ,'THE LOCATIONS ARE : '
    8     FORMAT(T2,A,/,T2,A,/T2,A)
          WRITE(6,9)(LOC(I),I=1,NLOC)
    9     FORMAT(/,T2,A30)
          READ(5,*) LOCATN
          GO TO 200
*
* IF THE LOCATION IS VALID, THEN CONFIRM IT IS THE DESIRED
* LOCATION. IF IT IS NOT THE DESIRED LOCATION, THEN GO TO
* 180 AND ASK FOR THE DESIRED LOCATION .
*
  210 ELSE
          WRITE(6,10)'THE LOCATION YOU WANT IS : ',LOC(LOCATN)
   10     FORMAT(T2,A,T29,A30)
          WRITE(6,11)'KEY IN 1 FOR YES, 2 FOR NO .'
  11   FORMAT(T2,A)
       READ(5,*) CORLOC
 220   IF(CORLOC.NE.1) GO TO 180
 230   CONTINUE
     ENDIF
 240 CONTINUE
*
* ASK THE USER TO ENTER THE DESIRED BASE TEMPERATURE .
*
 250 WRITE(6,12)'WHAT IS THE DESIRED BASE TEMPERATURE ?'
    +           ,' (TO ONE DECIMAL)'
  12 FORMAT(T2,A,T41,A)
 260 READ(5,*) BASET
*
* CONFIRM THAT THE ENTERED BASE TEMPERATURE IS THE ONE
* DESIRED .  IF IT IS NOT THE DESIRED TEMPERATURE, THEN ASK
* THE USER TO ENTER THE BASE TEMPERATURE AGAIN .
*
 270 WRITE(6,13)'THE BASE TEMP YOU DESIRE IS : ',BASET
  13 FORMAT(T2,A,T39,F7.1)
 280 WRITE(6,14)'KEY IN 1 FOR YES, 2 FOR NO .'
  14 FORMAT(T2,A)
 290 READ(5,*) CORT
 300 IF(CORT.NE.1) GO TO 250
 310 CONTINUE
*
* THE FOLLOWING LOOP CALCULATES THE NUMBER OF DEGREE-HOURS
* IN ANY ONE DAY .  IT THEN SUMS THE DAYS UP OVER THE MONTH
* TO GET THE NUMBER OF DEGREE-HOURS IN ANY PARTICULAR MONTH.
* THIS VALUE IS CONVERTED TO DEGREE-DAYS FOR THAT MONTH .
* THE OUTER LOOP CONTROLS WHICH MONTH IS BEING
* DEALT WITH.  THE INNER LOOP CONTROLS WHICH DAY OF MONTH
* IS BEING DEALT WITH. FIVE DIFFERENT CASES EXIST; HENCE THE
* USE OF THE IF-THEN-ELSEIF BLOCK .
*
 320 DO 410 M=1,NMON
       DAY=DAY+DAYSM(M)
       N=N+1
*
 330   DO 400 N=N,DAY
 340     IF((TMIN(N,LOCATN).GE.BASET).AND.
    +      (TMIN(N+1,LOCATN).GE.BASET)) THEN
           DHR1=0.0
           DHR2=0.0
           DHR(M)=DHR(M)+DHR1+DHR2
*
* CASE 2 OCCURS WHEN ONE OF THE MINIMUM TEMPERATURES IS
* LESS THAN THE BASET, ONE IS GREATER, AND THE MAXIMUM
* FOR THE DAY IS ALSO GREATER THAN THE BASET TEMPERATURE.
*
 350     ELSEIF((TMIN(N,LOCATN).GE.BASET).AND.
    +           (TMIN(N+1,LOCATN).LT.BASET).AND.
    +           (TMAX(N,LOCATN).GE.BASET)) THEN
           Y2M=(TMIN(N+1,LOCATN)-TMAX(N,LOCATN))/12.
```

```
        B2=TMAX(N,LOCATN)-(Y2M*12.)
        HR2=(BASET-B2)/Y2M
        DHR2=((24.-HR2)*(BASET-B2))+((Y2M/2.)*((HR2**2)
   +        -(24.**2)))
        DHR(M)=DHR(M)+DHR2
        HOURS=HOURS+(24.-HR2)
 360   ELSEIF((TMIN(N,LOCATN).LT.BASET).AND.
   +    (TMIN(N+1,LOCATN).GE.BASET).AND.
   +    (TMAX(N,LOCATN).GE.BASET)) THEN
        Y1M=(TMAX(N,LOCATN)-TMIN(N,LOCATN))/12.
        B1=TMAX(N,LOCATN)-(T1M*12.)
        HR1=(BASET-B1)/Y1M
        DHR1=(BASET-B1)*HR1-((Y1M/2.)*(HR1**2))
        DHR(M)=DHR(M)+DHR1
        HOURS=HOURS+(12-HR1)
*
* CASE 3 OCCURS WHEN BOTH OF THE MINIMUM TEMPERATURES
* AND THE MAXIMUM ARE BELOW THE BASE TEMPERATURE .
*
 370   ELSEIF((TMIN(N,LOCATN).LT.BASET).AND.
   +        (TMIN(N+1,LOCATN).LT.BASET).AND.
   +        (TMAX(N,LOCATN).LT.BASET)) THEN
        DHR1=0.5*12.*((BASET-TMIN(N,LOCATN))+(BASET
   +        -TMAX(N,LOCATN)))
        DHR2=0.5*12.*((BASET-TMAX(N,LOCATN))+(BASET
   +        -TMIN(N+1,LOCATN)))
        DHR(M)=DHR(M)+DHR1+DHR2
        HOURS=HOURS+24
*
* CASE 4 OCCURS WHEN BOTH MINIMUMS ARE BELOW THE BASE
* TEMPERATURE AND THE MAXIMUM TEMPERATURE IS ABOVE THE BASE
* TEMPERATURE .
*
 380   ELSEIF((TMIN(N,LOCATN).LT.BASET).AND.
   +        (TMIN(N+1,LOCATN).LT.BASET).AND.
   +        (TMAX(N,LOCATN).GE.BASET)) THEN
        Y1M=(TMAX(N,LOCATN)-TMIN(N,LOCATN))/12.
        B1=TMAX(N,LOCATN)-(Y1M*12.)
        HR1=(BASET-B1)/Y1M
        DHR1=(BASET-B1)*HR1-((Y1M/2.)*(HR1**2))
        Y2M=(TMIN(N+1,LOCATN)-TMAX(N,LOCATN))/12.
        B2=TMAX(N,LOCATN)-(Y2M*12.)
        HR2=(BASET-B2)/Y2M
        DHR2=((24.-HR2)*(BASET-B2))+((Y2M/2.)*((HR2**2)
   +        -(24.**2)))
        DHR(M)=DHR(M)+DHR1+DHR2
        HOURS=HOURS+(24-HR2)+(12-HR1)
*
* CASE 5 EXISTS IF FOR SOME REASON THE DATA DOES NOT FIT
* ANY OF THE OTHER CASES .
*
 390  ELSE
       WRITE(6,15)'DATA DOESN"T FIT TESTS. DATA IS FOR: '
   +          ,LOC(LOCATN),' ON ',N,' DAY OF THE YEAR.'
 15        FORMAT(T2,A,/,T2,A,T32,A,T36,I5,T41,A)
         ENDIF
400    CONTINUE
       N=N-1
410 TIME(M)=HOURS/24
*
* THE FOLLOWING LOOP CONVERTS DEGREE-HOURS TO DEGREE-DAYS.
*
420 DO 430 M=1,NMON
       DHR(M)=DHR(M)/24.0
       DHR(M)=DHR(M)*10.0
       DHR(M)=AINT(DHR(M))
       DHR(M)=DHR(M)/10.0
430 CONTINUE
*
* THIS IS THE OUTPUT SECTION .  THE RESULTS
* ARE PRINTED ON THE SCREEN AT THE TERMINAL.  THEY ARE ALSO
* STORED IN A TEMPORARY FILE CALLED -RES .
*
440 WRITE(6,16)'THE FOLLOWING RESULTS ARE FOR THE '
     +  ,LOC(LOCATN),' AREA. ','BASE TEMP = ',BASET
 16 FORMAT(T2,A,/,T2,A,T32,A,5X,A,1X,F6.1)
450 WRITE(6,17)
 17 FORMAT(/,'DEGREE DAYS',12X,'MONTH',5X,'DAYS')
460 WRITE(6,18)(DHR(M),MON(M),TIME(M),M=1,NMON)
 18 FORMAT( F9.2,12X,A12,2X,F7.1)
470 WRITE(7,16)'THE FOLLOWING RESULTS ARE FOR THE '
     + ,LOC(LOCATN),' AREA. ','BASE TEMP = ',BASET
480 WRITE(7,17)
490 WRITE(7,18)(DHR(M),MON(M),TIME(M),M=1,NMON)
500 WRITE(6,19)'RESULTS MAY BE VIEWED IN THE TEMPORARY'
     +          ,' FILE -RES .  TO GET A HARD COPY, COPY'
     +          ,' -RES TO THE PRINTER .'
 19 FORMAT(T2,A,/,T2,A,/,T2,A)
*
510 WRITE(6,20) 'WOULD YOU LIKE TO MAKE ANOTHER RUN ?'
     +  ,'KEY IN 1 FOR YES , 2 FOR NO .'
 20 FORMAT(T2,A,/,T2,A)
520 READ(5,*) RUN
*
530 IF(RUN.EQ.1) THEN
       I=0
       J=0
       K=0
       M=0
       N=0
       HOURS=0.
       DAY=0
       DO 535 I=1,12
         DHR(I)=0.0
         TIME(I)=0.
535      CONTINUE
       GOTO 180
    ELSE
```

```
        GOTO 540
      ENDIF
      CONTINUE
  540 STOP
*
      END
```

14. APPENDIX D: COMPUTER PROGRAMS - DATA CONVERSIONS

```
C  Program: CRUNCH

C  The following program turns the output from the data
C logger into sensible numbers.

C  PROGRAMMING LANGUAGE: FORTRAN

C  The output from the data-logger was in the form of the
C   time followed by 29 digitized voltages representing
C   thermistor readings. Each set of thermistor readings
C   was separated by 5 pressure and dewpoint readings.

C   VOLT - array containing the raw theristor voltages
C   TEMP - array of the converted temperatures
C   PRESS - pressure readings
C   DEW - the dewpoint readings

C   Output devices:
C           6 - file containing temperatures in an easily
C               read form
C           7 - file containing temperatures to be read by
C               other programs
C           8 - file of the dewpoints
C           9 - file of the pressures

      REAL VOLT(29),TEMP(29),PRESS(14),DEW(14)
        ISEQ=1

C  Titles for output files

      WRITE (8,998)(I,I=1,7)
      WRITE (9,999)(I,I=1,7)

C  Read the time and the thermistor voltages

 25   READ(5,90) TIME,(VOLT(I),I=1,29)
      IF (TIME .EQ. 1.111) GOTO 26
       TIME=TIME*10

C  Convert the voltages into temperatures

       DO 45 K=1,29
       T=VOLT(K)
       R=T*2000.-1.2
       IF(R.LE.0.)R=1000.
       R=ALOG(R)
       R=.209099055D-6*R*R*R+.2758512252D-3*R+.1379588514D-2
 45    TEMP(K)=1.0/R-273.16

C   Output the converted tempeatures
```

```
        WRITE (6,1000) TIME,TEMP(1),TEMP(26),TEMP(28)
     + ,TEMP(29),TEMP(14),TEMP(15),TEMP(27)
        WRITE (6,1001) TEMP(10),TEMP(11),TEMP(12),TEMP(13)
        WRITE (6,1002) (TEMP(II),II=2,8,2),
     + (TEMP(LL),LL=3,9,2),TEMP(16),TEMP(18),TEMP(22),
     + TEMP(24),TEMP(17),TEMP(19),TEMP(23),TEMP(25)

C    Read the pressure and dewpoint voltages and convert them

        DO 34 N=1,5
           READ(5,90) A,B
           DEWP=13.6*A-28.6
            IF (DEWP .LE. 0.) DEWP=1.1064*DEWP - 0.066111
           DEW(ISEQ)=DEWP
           PRESS(ISEQ)=B*.2*25.4
           ISEQ=ISEQ+1

C     Every 15th reading is meaningless

 33         IF (ISEQ .GT.14) GOTO 51
 34     CONTINUE
        GOTO 52

C     Output the values

 51       WRITE (8,997)TIME,(DEW(IJ),IJ=1,14)
          WRITE (9,997)TIME,(PRESS(JI),JI=1,14)
          ISEQ=1
          READ(5,90)A,B
          GOTO 25
 52     WRITE (7,91)TIME,(TEMP(III),III=1,19),
     +(TEMP(I2),I2=20,29)
        GOTO 25
 90     FORMAT ( 16F7.5)
 91     FORMAT ( 15F8.2)
  997 FORMAT ( F8.2,7F10.1,/,8X,7F10.1 )
  998 FORMAT ( 10X,'DEW POINTS  deg C',/,'Time      '
     +,7I10,//)
  999 FORMAT ( 10X,'PRESSURES  mm H20',/,'Time      '
     +,7I10,//)
 1000 FORMAT( /,'Time is',F8.5,3X,'OUT TEMP:',4F8.2,/,
     +'IN TEMP',3F8.2)
 1001 FORMAT( 'Before supply fan',2F8.2,'C','  After',2F8.2)
 1002 FORMAT( 'SUPPLY PROFILE:'4F8.2,/,13X,4F8.2,/,
     +'EXHAUST PROFILE:'4F8.2,/,16X,4F8.2)
 26     STOP
        END
```

```
1 ' PROGRAMMING LANGUAGE: BASIC
2 ' This program allows the output from the particle
3 '  counter to be put into an easily read format. The
4 '  data is a series representing the particles per 0.01
5 '  cu.ft. and arranged as follows: (sizes in micro-meters)
6 '  0.5 and larger, 1.0 and larger, 2.0 and larger, 5.0 and
7 '  larger, and 10 and larger. In addition to the dust, the
8 '  input contains the time, partial readings, and numerous
9 '  zeros which have to be sorted out.
10 '
15 DIM CELL(5)                      ' array for reading data
16 INPUT "RUN?"; RN                 ' which run is data for?
17 OPEN "B:DST" FOR APPEND AS #2    ' output file
18 PRINT #2,"RUN" RN
19 OPEN "B:DUST" FOR INPUT AS #1 " ' input file
20 INPUT #1, A                      ' read a value
21 IF A = 0 GOTO 20                 ' ignore if it is a zero
22 PRINT A                          ' show the value
23 INPUT "0=GO 1=TIME"; DOD  ' is value a count or the time?
24 IF DOD = 0 GOTO 29               ' if a count, go to 29
25 TIME=A                           ' set the time
26 GOTO 20
27 CELL(1)=A          ' put the count in cell number one
28                    ' and read 4 more
30 INPUT #1, CELL(2), CELL(3), CELL(4), CELL(5)
31                    ' show the five values
35 PRINT CELL(1), CELL(2), CELL(3), CELL(4), CELL(5)
36             ' decide if the values are particle counts
37             ' or meaningless
40 PRINT "1=CHANGE 0=PRINT 2=READ MORE"
41 INPUT CHG
42 IF CHG = 0 GOTO 56   ' data okay
43 IF CHG = 2 GOTO 90   ' get more values
50 '
51 INPUT "1"; CELL(1)   ' change values if most values seem
52 INPUT "2"; CELL(2)   ' reasonable except for noise or
53 INPUT "3"; CELL(3)   ' missing value, etc.
54 INPUT "4"; CELL(4)   ' then output the result
55 INPUT "5"; CELL(5)
56 PRINT #2, TIME CELL(1) CELL(2) CELL(3) CELL(4) CELL(5)
57 GOTO 20              ' get next value
58 '
59                ' get more values to compare to first set
90 INPUT #1, CELL(6),CELL(7), CELL(8), CELL(9), CELL(10)
91 PRINT CELL(6)  CELL(7)  CELL(8)  CELL(9)  CELL(10)
93 GOTO 50
```

15. APPENDIX E: COMPUTER PROGRAMS - HEAT EXCHANGER PERFORMANCE CALCULATIONS

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1 ' Program: HEAT      Language: BASIC

2 ' The following program calculates the heat transfer
3 ' coefficients for the 3 regions of the heat exchanger
4 ' using the temperatures produced by the program CRUNCH.

6 DIM TEMP(30)                        ' array used to read data
7 OPEN "B:HEAT" FOR APPEND AS #2      ' output file: heat
8 PRINT #2,"HEAT TRANSFER COEFFICIENTS IN W/m²°C"
9 PRINT #2,"RUN  POSITION 1  2   3"
10 OPEN "B:??????" FOR INPUT AS #1 ' specify input file
11 PRINT "SUPPLY flow rate L/s?"
12 INPUT SUPP                         ' input the supply flow
13                                    ' rate for the run
14 SUPP=.001*SUPP/11.16               ' flow rate for 1 channel
18 SUPP=SUPP/.8                       ' convert to mass flow

20 A=0                                ' initialize variables
25 SUM1=0
25 SUM2=0
25 SUM3=0
26 SMQ1=0
26 SMQ2=0
26 SMQ3=0

27 K=0                                ' read one set of data
28 K=K+1
29 INPUT #1,TEMP(K)
30 IF K<30 GOTO 28

31  ' The temperatures along the heat exchanger were
32  ' measured at 4 locations (A through D; A at inlet,
33  ' D at outlet). Each location had 2 thermistors
33  ' This divided the core into 3 regions.
34  ' Lines 51-58 calculate the temperature of the supply
35  ' air at each thermistor location (TA - TD) and of the
36  ' exhaust air for each region (T1E ,T2E, T3E).

51 A=A+1
52 TA=(TEMP(3)+TEMP(4))/2
53 TD=(TEMP(9)+TEMP(10))/2
54 T1E=(TEMP(17)+TEMP(18)+TEMP(19)+TEMP(20))/4
55 T2E=(TEMP(23)+TEMP(24)+TEMP(19)+TEMP(20))/4
56 T3E=(TEMP(23)+TEMP(24)+TEMP(25)+TEMP(26))/4
57 TB=(TEMP(5)+TEMP(6))/2
58 TC=(TEMP(7)+TEMP(8))/2
59 HEAT1=(T2-T1)*SUPP*1006            ' Rate of heat transfer,
60 HEAT2=(T3-T2)*SUPP*1006            ' (W) for[illegible]ch region.
61 HEAT3=(T4-T3)*SUPP*1006
62 HC1=HEAT1/.552/(T1E-(TB+TA)/2)  ' Calculate the heat
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63 HC2=HEAT2/.268/(T2E-(TB+TC)/2)   ' transfer coefficients
64 HC3=HEAT3/.522/(T3E-(TC+TD)/2)   ' for each region and
71 PRINT #2,  HC1 HC2 HC3           ' print in output file

72 SUM1=SUM1+HC1                    ' sum for calculating
73 SUM2=SUM2+HC2                    ' average and standard
75 SUM3=SUM3+HC3                    ' deviation
76 SMQ1=HC1**2+SMQ1
77 SMQ2=HC2**2+SMQ2
78 SMQ3=HC3**2+SMQ3
80 GOTO 27                          ' get next set

100 AVE1=SUM1/A                     ' at end of data
101 AVE2=SUM2/A                     ' GOTO 100 for averages
103 AVE3=SUM3/A                     ' and standard deviations
110 SD1=(SMQ1-(A*AVE1**2))/(A-1)
111 SD2=(SMQ2-(A*AVE2**2))/(A-1)
112 SD3=(SMQ3-(A*AVE3**2))/(A-1)
120 S1=SD1**.5
121 S2=SD2**.5
122 S3=SD3**.5
150 PRINT "H1="AVE1 "SD="S1
151 PRINT "H2="AVE2 "SD="S2
152 PRINT "H3="AVE3 "SD="S3
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1 ' Program: PORTION    Language: BASIC

2 ' The following program is similar to HEAT except that
3 ' it finds the portion of the total heat that is
4 ' tranferred by each region.

5 DIM TEMP(30)
6 OPEN PIRT FOR APPEND AS #2        ' output file
7 PRINT #2, "PORTION OF HEAT TRANSFERED"
8 PRINT #2,"RUN  POSITION 1  2   3"
10 OPEN "B:??????" FOR INPUT AS #1 ' specify input file
20 A=0
25 SUM1=0
25 SUM2=0
25 SUM3=0
26 SMQ1=0
26 SMQ2=0
26 SMQ3=0
27 K=0
28 K=K+1
29 INPUT #1,TEMP(K)
30 IF K<30 GOTO 28

31 ' The calculations are simpler for this program
32 ' since the heat transferred is proportional to the
33 ' rise in the supply air temperature.

51 A=A+1
52 TA=(TEMP(3)+TEMP(4))/2
53 TD=(TEMP(9)+TEMP(10))/2
57 TB=(TEMP(5)+TEMP(6))/2
58 TC=(TEMP(7)+TEMP(8))/2

59 T1=(TB-TA)                          ' temperature changes
60 T2=(TC-TB)                          ' by region
61 T3=(TD-TC)
65 TOTAL=(TD-TA)                       ' total change

66 H1=T1/TOTAL                         ' fraction by region
67 H2=T2/TOTAL
68 H3=T3/TOTAL
69 PRINT #2, H1 H2 H3               ' output results
72 SUM1=SUM1+H1
73 SUM2=SUM2+H2
75 SUM3=SUM3+H3
76 SMQ1=H1**2+SMQ1
77 SMQ2=H2**2+SMQ2
78 SMQ3=H3**2+SMQ3
80 GOTO 27
100 AVE1=SUM1/A
101 AVE2=SUM2/A
103 AVE3=SUM3/A
110 SD1=(SMQ1-(A*AVE1**2))/(A-1)
111 SD2=(SMQ2-(A*AVE2**2))/(A-1)
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112 SD3=(SMQ3-(A*AVE3**2))/(A-1)
120 S1=SD1**.5
121 S2=SD2**.5
122 S3=SD3**.5
150 PRINT "H1="AVE1 "SD="S1
151 PRINT "H2="AVE2 "SD="S2
152 PRINT "H3="AVE3 "SD="S3
```